STEAM PIPES

STEAM PIPES:

THEIR DESIGN AND CONSTRUCTION

A Treatise of the Principles of Steam Conveyance and Means and Materials Employed in Practice, to Secure Economy Efficiency and Safety

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PREFACE

IF any engineer will refer to his various text-books for information on steam piping, he will find very little to assist him. The earliest steam pipes served to convey steam at atmospheric pressure into cylinders of large size. The object of the steam was simply to fill the cylinder with a condensible vapour, so that upon cooling it there was a vacuum formed and the air pressure on the opposite face of the piston was made to do work. So long as the piston could be drawn up by the weight of the pump spear rods hung to the opposite end of the beam, it mattered little that the pipes were small, for there was no object in filling the cylinder with steam at full boiler pressure, for the vacuum obtained on condensing would be better as the steam was the less in quantity. When the steam engine was made rotative its piston speed became great and more regular, and pipes of a definite size began to be found necessary. Most of the early books on the steam engine gave engine proportions of a very empirical order, deducing them from the cylinder diameter by formulæ that were sound only so far as they were applied to engines constructed

to standard practice. Standard practice in so far as beam engines were concerned was very much as Boulton and Watt had made it, and no doubt the proportions given by these curious old formulæ were properly deduced from what experience had fixed upon as good practice. Our modern practice in steam pipes is thus the outgrowth of Boulton and Watts' early experience. It is a well-known fact that when steam escapes to atmosphere through an opening, the weight of flow is proportionate approximately to the absolute pressure. Thus steam at ninety pounds absolute pressure will escape three times as fast as regards weight as will steam of thirty pounds absolute pressure. Yet as steam pressures have advanced the area of pipes has not diminished accordingly. Any steam boiler, no matter at what pressure it may be worked, will convert a steady weight of water into steam. It follows therefore that for boilers of high pressure the steam pipes may be proportionately less than for low pressures. But as all practice has been towards higher pressures, it has always been the case that steam pipe mounting blocks, steam entrance pipes to engines, steam valves for any stated size of boiler, have been gauged by last year's practice rather than this year's, and, as is always the case in engineering practice, there has been a delay in cutting down pipe dimensions to the equivalent of the rise in pressure. While steam pipes may be too small they are probably more often too large, and not the less so that they have been called frequently

to supply many engines of high rotative speed which make a demand on the steam supply that is practically continuous. Apart from mere sufficiency of piping, as such, there are the numerous details involved in flange diameters, bolts, centre lines, sockets, joints, and, not less important, valves, which all demand investigation. It is the object of this book to bring together such information as may be useful in connection with steam pipes as will be of assistance to the engineer who has to face problems of this sort. Steam piping to-day is so costly an item, especially when large valves are employed, that every effort should be made to minimize such cost without sacrifice of efficiency. No attempt has been made to put together everything that is published on steam piping. Selections only have been possible, and the author is indebted among others to the Babcock & Wilcox Co. for permission to reproduce from their book *Steam* and other of their pamphlets; to Mr. A. J. Lawson, of the British Electric Traction Co.; the Mannesmann Tube Co.; the Cruse Superheater Co.; and to Mr. Arthur Venning; also to Messrs. Holden & Brooke; Messrs. Templer & Ranoe, whose productions the author has employed to illustrate the book; Messrs. Yates & Thom, for numerous illustrations of Lancashire practice; Mr. Stromeyer, of the Manchester Steam Users' Association; Messrs. J. E. & S. Spencer, Ltd.; Mr. Thos. Walker, of Tewkesbury, and others.

In the book will be found various tables of dimensions of junction pieces, valves and flanges. In

printing these the author has merely been influenced by a desire to put before readers a few examples as a guide, and not as a fixed and determined standard. There are many so-called standards which differ from one another in but little, and perhaps the most important detail to standardize is the flange as regards diameter, bolt circle and bolt numbers, and their relation to centre lines. A committee is now sitting on this subject, and doubtless will arrive at a result which engineers can accept. It will probably be useful as a guide to have general dimensions of even a single make of valve, but the author would suggest that overall dimensions of valve bodies ought also to be made standard, for such would make it possible to get out a whole system of pipes before deciding on where the valves should be obtained.

There are so few and so small differences between one maker's products and those of another that a universal standard should be quite practicable.

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CHAPTER I

Steam Pipes: Their Duty and Object

THE object of a steam pipe is to convey steam from point to point.

A steam pipe as ordinarily understood is for the conveyance of steam from the boiler to the engine, or to other apparatus, such as a dye vat or brewing copper, etc.

In the early days of steam engineering there were no steam pipes ; the working cylinder of the engine was wholly or partially connected to the boiler or joined by a narrow neck, which, becoming gradually longer, developed into a pipe. Sometimes of rectangular section for convenience under special circumstances, the natural and usual cross-section of a pipe is the circle, that being a figure which contains a maximum of area within a minimum circumscribing boundary, and also being the only figure of maximum strength to resist bursting, and therefore requiring no internal or other stays.

Since a steam engine gives the best results at the highest pressures, the duty of a steam pipe is to convey steam to the engine with a minimum of loss of pressure. Steam being hotter than the air surrounding the pipes must lose some of its heat in its passage

through pipes. Obviously, therefore, pipes must be of minimum size. This is inconsistent with a minimum pressure loss, and a compromise must therefore be come to between loss of heat and loss of pressure. If that compromise could be worked out, it would be found by equating the loss of coal due to loss of economy consequent on loss of a given amount of pressure, and the loss by radiation of heat that would be incurred by making the pipe large enough to prevent said loss of pressure.

Beyond a certain small loss of pressure, any further increase of pipe diameter affords so little further reduction of friction and adds so much to the heat radiation losses from the pipe surface that very large pipes must not be used, for they involve also increased capital expenditure in pipe sizes, coverings, flanges and valves out of all proportion to the small gain of pressure.

The broad principles to be observed to enable a pipe to perform a maximum duty are that it shall take as direct a course as practicable between any two points, shall be as smooth internally as it can reasonably be made, and shall have bends of large radius. All these points are compelled to be neglected by circumstances, but they afford a basis for design.

As a steam pipe is meant to convey steam and will be called on to convey as much water as may be formed in it by condensation due to cooling, this must be provided against by suitably covering the pipes with a non-heat conducting substance. Other-

wise not only is heat lost, but, water being formed, must be impelled along the pipe at the expense of the steam, which will lose pressure as a result.

As a steam pipe failure will cause the stoppage of a whole power station, the practice has grown up among electrical engineers of duplicating the steam pipes. Hence arose that nuisance the ring main, with its myriad of costly valves, its maximum of condensation and its minimum of safety. The ring main ought only to be employed where other conditions render it obligatory, and these only occur as a rule with initially bad designs. The ring main is not a steam engineer's device. Steam engineers have always arrived at directness of pipes knowing the losses of heat in long pipes, and they insist on the use of the best materials so as to minimize the chance of failure rather than countenance the duplication of bad work which has arisen from want of knowledge of steam engineering conditions and an apparently foregone conclusion that break-downs are necessary, and must be encouraged to occur by provision of a maximum of parts to fail.

CHAPTER II

The Flow of Steam

RANKINE, in his work on the Steam Engine, gives the following formula for the velocity of flow of steam where—

V = velocity in feet per second.
g = gravity = 32·2.
γ = 1·3.
p_0 = ideal pressure at 32° F.
T_0 = absolute temp. at 32° F.
T_1 = ,, ,, at pressure p_1.
T_2 = ,, ,, ,, ,, p_2.
p_1 = ,, pressure in boiler.
p_2 = ,, ,, at steam chest.
v_0 = volume ideal at 32°.
$p_0 v_0$ = 42141.
k = coefficient of contraction.
v_1 = volume of 1 pound of steam at p_1.
v_2 = ,, ,, 1 ,, ,, ,, ,, p_2.

$$V = \sqrt{\left\{\frac{2 g \gamma}{\gamma - 1} \, \frac{p_0 v_0 T_1}{T_0} \left(1 - \left(\frac{p_2}{p_1}\right) \frac{\gamma - 1}{\gamma} \right)\right\}} \quad (a)$$

Substituting values this becomes—

$$V = \sqrt{\left\{\frac{64{\cdot}4 \times 1{\cdot}3 \times 42141 \times 875}{493{\cdot} \times 0{\cdot}3} \times 0{\cdot}0035\right.}$$

$$= \sqrt{71066} = 265$$

feet per second, where p_1 = 200 lb. absolute and

$p_2 = 197$ lb. That is to say, with a drop of pressure of 3 pounds the velocity in a short straight pipe may be 265 feet per second.

For small differences of pressure Rankine gives an approximate formula—

$$V = \sqrt{\frac{2g^\circ \times 42141 \cdot \; T_1 \; (p_1 - p_2)}{T_0 \quad p_1}} \quad (b)$$

Substituting values gives $V = \sqrt{72260} = 270$ feet per second, which is not a serious difference from the complicated formula.

In his Rules and Tables, Rankine gives a rough approximation of the weight of steam flow per second where the initial and final pressures are p_1 and p_2 respectively, and q = pounds per second per unit of area.

(1) Where $p_2 =$ or $< \frac{3}{5} p_1$; $q = p_1 \div 70$ nearly.

This formula is only useful when the external or final pressure is low.

(2) Where $p_2 > \frac{3}{5} p_1$, $q = \frac{p_2}{42} \sqrt{\frac{p_1 - p_2}{\frac{2}{3} p_2}}$

Applying this formula (2) to the case of steam of 200 lb. (28,800 lb. per square foot) flowing with a loss of 3 lb., or to a final pressure of 197 lb. (28,368 lb. per square foot), we have by (2)—

$$q = \frac{28368}{42} \sqrt{\frac{432}{18912}}, \text{ or } 675{\cdot}4 \sqrt{0{\cdot}02284},$$

or $675{\cdot}4 \times 0{\cdot}1511 = 102{\cdot}06$ pounds.

As at 197 pounds pressure there are 2·26 cubic feet per pound, the velocity of flow per second wil

be $2 \cdot 26 \times 102 = 230 \cdot 6$ feet, which again is not very seriously different from the velocities found by the more strict formulæ.

Ordinarily gases flow by virtue of the same rules as apply with liquids. The rule for the flow of a liquid is $v = \sqrt{2gh}$ or $v = 8\sqrt{h}$ (3), where h is the head in feet. For gases h is that height of a column of gas equivalent to the pressure. Then in the case in point of steam flowing from 200 lb. to a pressure of 197 lb., the pressure difference per square foot is 432 lb. and the mean density is 2·275 cubic feet per pound, whence the virtual head in feet is $h = 432 \times 2 \cdot 275 = 982 \cdot 8$ feet.

Then $v = 8\sqrt{982 \cdot 8} = 252$ feet per second.

It is obvious, therefore, that so far as regards steam velocity in ordinary practice no specially accurate formula is needed. Had the pressure difference been only one pound per square inch or 144 pounds per square foot, the velocity would have been $v = 8\sqrt{\frac{982 \cdot 8}{3}}$, or 144·8 feet per second.

In Spon's *Dictionary of Engineering* the following metrical formula is given—

$$v = \sqrt{2\,g\,(P - p)\,\frac{d^1}{d}} \text{ where} \quad (4)$$

v is the velocity in metres per second.

g° = gravity = 9·8088 metres per second.

P and p = initial and final pressures in metres of mercury column.

d^1 = Sp. gr. of mercury.

d = ,, ,, ,, steam.

This formula reduces to $v = \sqrt{\frac{202 \cdot 74 (P-p)}{d}}$, when P and p are the pressures stated in atmospheres.

Our standard example works out as $P - p = 0 \cdot 203$, and $v = \sqrt{5837} = 76 \cdot 00$ metres, or 249 feet per second as found by the previous rule.

The calculated velocities above found cannot be employed in practice, because they are reduced by friction and by condensation which produces friction.

This is one reason why superheated steam travels so much better than wet steam. The velocity to be counted upon must therefore be reduced in accordance with the length of pipe, its bends and other resistances.

D'Aubisson found that resistance is directly proportional to length, that it increases with the square of the velocity, and it is inversely as the diameter.

He applied a correction to the velocity found by the metrical formula (4) namely, $\sqrt{\frac{D}{(0 \cdot 0238\,L + D)}}$ where D and L are the diameter and the length of the pipe in metres. D'Aubisson's experiments were made with air in tin pipes, and the formula will be the more correct as the pipes are smoother. Probably for cast-iron pipes the reduction of flow will exceed that given in the formula worked out for a pipe 30 metres long and 0·25 diameter, which corresponds with a pipe about 100 feet by 10 inches diameter ; the result is $\sqrt{\frac{0 \cdot 25}{0 \cdot 964}}$, or practically the

velocity is reduced to one half, showing that friction is very considerable.

It is usual in practice to require to know the diameter where the quantity and the fall of pressure are given as in fixing the pipe diameter for a boiler of a given evaporative capacity at a certain distance from the boiler.

Calling Q the volume in cubic metres per second.

L = pipe length in metres.

D = pipe diameter in metres.

P = difference of pressure available in metres of mercury.

d = density of gas relative to water.

Then $$D = \cdot 36 \sqrt{\frac{(0\cdot 0238\, L + D) \times Q^2\, d}{P}}$$

This formula may first be worked out with an assumed value for the D under the root sign. It may be taken conveniently as that diameter which will require a flow of 30 metres per second. This assumed value is then used under the root sign and the formula worked out. If the calculated and the assumed values of D are found to differ, the new value of D as calculated can now be used under the root sign and a fresh value calculated which will be very close to the first calculated value where pipes are of usual lengths or several diameters long. For convenience in using these metrical formulæ the following equivalents will be useful in reducing British data to metrical—

1 metre = 39·37 inches = 3·28 feet.

1 foot = 0·305 metre.

1 inch = 0·0254 metre
1 cubic foot = 0·0283 cubic metre.
1 ,, metre = 35·316 cubic feet.
1 pound per square inch difference of pressure = 0·052 metre of mercury column.

A formula sometimes employed for the velocity of flow in a pipe is $V = 50\sqrt{\frac{H\,D}{L}}$; the value of H being $v.p.$ 144, where V = feet per second, v the volume in cubic feet of 1 pound of steam at the initial pressure, L and D the length and diameter of the pipe in feet, and p the difference of pressure between the two ends of the pipe. The number of cubic feet per second is then Va, where a is the pipe area in square feet. Thus for a 6-inch pipe 50 feet long, carrying steam of 100 pounds absolute pressure, with a drop of 5 pounds, we have $v = 4{\cdot}29$ from any steam table, whence $H = 4{\cdot}29 \times 5 \times 144 = 3088{\cdot}8$. Hence, $V = 50\sqrt{\frac{3088{\cdot}8 \times 0{\cdot}5}{50}} = 278$ feet per second, and $Va = 278 \times 0{\cdot}196 = 54{\cdot}5$ cubic feet of flow per second.

Hutton gives a rule for the outflow of steam into an external pressure not more than 58 per cent. of the internal pressure, as follows:—

W = weight of steam discharged per minute.

Then $W = \frac{P.A}{C}$, where

P = absolute pressure in pounds per square inch.
A = area of pipe in square inches.
C = 1·38 for pipes up to 10 ft. long.

= 1·39 for pipes up to 40 feet long.
= 1·42 „ „ „ „ 70 „ „
= 1·45 „ „ „ „ 100 „ „

P of course must not be less than 12 pounds above the atmosphere.

The velocity of flow through a valve or short pipe is $V = 32 \sqrt{T + 460}$, when T is the temperature and V = feet per second.

Neither of these rules bears upon boiler steam pipes, because a drop of 42 per cent. or more could not be tolerated. The rules are useful for special cases.

Practical Rules.

Hutton gives as a good practical rule that in ordinary cases steam velocity should not exceed 85 feet per second = 5,100 feet per minute. If very short and straight the velocity may be 110 feet. He supposes steam to follow the piston for full stroke, and gives the following rule for steam pipe area—

$$\frac{\text{Cylinder Area} \times \text{Piston Speed per minute}}{5100} = \text{Pipe area.}$$

This rule is purely empirical, for it supposes continuous steaming and neglects the higher piston velocity at middle stroke.

Referred to evaporation the steam pipe area is given as—

$$A = \frac{\text{Pounds of Steam per minute} \times \text{volume of steam relative to water}}{\text{Velocity in feet per minute} \times 62{\cdot}42 \div 144.}$$

Mr. Stromeyer, of the Manchester Steam Users' Association, gives as a rough rule for sectional area of steam pipes in square inches = A.

$$A = \frac{180 \times \text{Sum of widths of furnaces in inches}}{\text{Absolute Pressure.}}$$

This rule corresponds with a velocity of 8,000 feet per minute and a fuel consumption at the rate of 25 pounds per square foot per hour on grates 6 feet long. Obviously the rule has been empiricised from ordinary rates of evaporation and combustion.

In ordinary land practice the constant becomes 240, but where there is an ample excess of boiler pressure, as in case of Belleville boilers, for example, which carry pressures much in excess of what is permitted to reach the engines, the constant need not be more than 120.

The combined formula for flow of steam due to difference of pressure and length of pipe and diameter is—

$$W = 87\sqrt{\frac{D\,(p_1 - p_2)\,d^5}{L\left(I + \frac{3 \cdot 6}{d}\right)}}, \quad \text{where} \quad (1)$$

W = the weight in pounds per minute.

D = the weight per cubic foot of steam at the pressure.

p_1 and p_2 = the initial and final pressures.

L = the pipe length in feet.

d = the pipe diameter in inches.

Table I has been calculated for *Steam* (B. & W. Boiler Co.) from this formula for pipes having a length of 240 diameters, straight and smooth. The results are in pounds per minute. In using this formula it must be noted that actual pipe diameters are employed as per Table II., which gives the pipe diameters on which the Table I. is calculated for all sizes below six inches.

TABLE I. OF FLOW OF STEAM THROUGH PIPES.

Initial Pressure by Gauge. lb. per sq. in.	Diameter of Pipe in Inches. Length of each = 240 Diameters.													
	$\frac{3}{4}$	1	$1\frac{1}{2}$	2	$2\frac{1}{2}$	3	4	5	6	8	10	12	15	18
	Weight of Steam per Minute in Pounds, with One Pound Loss of Pressure.													
1 .	1·16	2·07	5·7	10·27	15·45	25·38	46·85	77·3	115·9	211·4	341·4	502·4	804	1177
10 .	1·44	2·57	7·1	12·72	19·15	31·45	58·05	95·8	143·6	262·0	422·7	622·5	996	1458
20 .	1·70	3·02	8·3	14·94	22·49	36·94	68·20	112·6	168·7	307·8	496·5	731·3	1170	1713
30 .	1·91	3·40	9·4	16·84	25·35	41·63	76·84	126·9	190·1	346·8	559·5	824·1	1318	1930
40 .	2·10	3·74	10·3	18·51	27·87	45·77	84·49	139·5	209·0	381·3	615·3	906·0	1450	2122
50 .	2·27	4·04	11·2	20·01	30·13	49·48	91·34	150·8	226·0	412·2	665·0	979·5	1567	2294
60 .	2·43	4·32	11·9	21·38	32·19	52·87	97·60	161·1	241·5	440·5	710·6	1046·7	1675	2451
70 .	2·57	4·58	12·6	22·65	34·10	56·00	103·37	170·7	255·8	466·5	752·7	1108·5	1774	2596
80 .	2·71	4·82	13·3	23·82	35·87	58·91	108·74	179·5	269·0	490·7	791·7	1166·1	1866	2731
90 .	2·83	5·04	13·9	24·92	37·52	61·62	113·74	187·8	281·4	513·3	828·1	1219·8	1951	2856
100 .	2·95	5·25	14·5	25·96	39·07	64·18	118·47	195·6	293·1	534·6	862·6	1270·1	2032	2975
120 .	3·16	5·63	15·5	27·85	41·93	68·87	127·12	209·9	314·5	573·7	925·6	1363·3	2181	3193
150 .	3·45	6·14	17·0	30·37	45·72	75·09	138·61	228·8	343·0	625·5	1009·2	1486·5	2378	3481

In order to render the table suitable for pipes with bends and valves, the values of these are as follows.

The resistance of the first opening from the boiler and that of a globe valve are each equal to a length of pipe $= \dfrac{114}{I + \dfrac{3 \cdot 6}{d}}$. This length works out as follows for various pipes—

in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.
$d = \frac{3}{4}$	1	$1\frac{1}{2}$	2	$2\frac{1}{2}$	3	4	5	6	8	10	12	15	18
$L = 20$	25	34	41	47	52	60	66	71	79	84	88	92	95

An elbow is equivalent to $\frac{2}{3}$ of a globe valve. These equivalent lengths are all added to the straight pipe length, and the total is the equivalent straight pipe.

Thus a 4-inch pipe 40 feet long (120 diameters) with a globe valve and three elbows, and the opening from the boiler is equivalent to a length of $120 + 60 + 60 + (3 \times 40) = 360$ diameters, or $1\frac{1}{2}$ times the tabular length. The flow through this pipe will be that given in the table multiplied by $1 \div \sqrt{1 \cdot 5}$, or 81·6 per cent. of the tabular number.

That is, for any length of pipe other than 240 diameters, divide 240 by the equivalent length, and take the square root of the quotient, which divide into the tabular weight. The result is the weight of flow for the new length.

Again, for any loss of pressure other than 1 pound, multiply the tabular figure by the square root of

the pressure drop. Thus a drop of four pounds instead of one pound should double the output.

The formula (1) is sometimes written—

$$W = 303 \cdot 25 \quad d^2 \sqrt{\frac{D\,(p_1 - p_2)}{L\left(1 + \frac{3 \cdot 6}{d}\right)}};$$ where L is the number of times the length is of the diameter. Obviously this brings the term d into the denominator, and enables the d^5 to come from under the root sign, for $d^2 = \sqrt{\frac{d^5}{d}}$.

The number 303·25 is the 87 of the previous formula multiplied by $\sqrt{12}$, which is necessary where it is changed to feet instead of being a multiple of a diameter in inches.

When steam flows from one pressure to any other pressure less than three-fifths of the initial pressure its velocity has the constant value 888 feet per second, so that the weight discharged varies with the density. Hence the rule for weight of outflow per minute W pounds.

W = area of opening $a \times 370 \times$ weight of a cubic foot of steam.

Rankine's formula is $W = \frac{6\,a\,p}{7}$, where a is the area in square inches and p is the absolute pressure. A coefficient of reduction $k = 0 \cdot 93$ is employed for a short pipe and $k = 0 \cdot 63$ for an opening in a thin plate, as a safety valve for example. When the steam flows into a pressure more than two-thirds the initial the formula becomes—

$$W = 1{\cdot}9\, a\, k \sqrt{(p - d)\, d},$$

where d is the difference of pressure. The result is substantially what all other formulæ give.

In a system of pipes in order that a correct balance may be found the proper proportion of any size of pipe to allot as an equivalent of any other size must be found. Pipes deliver according to the square of their diameters, but the same head will not produce the same velocity of flow in four 5-inch pipes as in their equivalent one 10-inch pipe.

The relative flow W in different pipes varies as $\sqrt{\dfrac{d^6}{d+3{\cdot}6}}$ where W = weight of fluid and d = diameter in inches.

In Table II., from *Steam* (Babcock & Wilcox Co.) the true or standard diameters of pipes are given, and in Table III. are given the equivalents of pipes in terms of other pipes. That part of the table above the diagonal line refers to standard pipes of the nominal diameter only. Below the diagonal the pipes are actually of the given diameter.

Thus below the top line 7, along the line 2, we find 29, or the number of nominal 2-inch pipes equal to a single 7-inch nominal, or again, 6·21 pipes of standard 7-inch size = 1 pipe of actual 14-inch size, but it requires 6·45 pipes of 7-inch standard to equal one standard 14-inch.

The table is useful, but it is calculated for American pipes and must be used with discretion with English pipes, though no very serious discrepancy will arise.

TABLE II. OF STANDARD SIZES, STEAM AND GAS PIPES.

Size, Ins.	Diameter. Inter-nal.	Diameter. Exter-nal.	Size, Ins.	Diameter. Inter-nal.	Diameter. Exter-nal.	Size, Ins.	Diameter. Inter-nal.	Diameter. Exter-nal.
$\frac{1}{8}$	·27	·40	$2\frac{1}{2}$	2·47	2·87	9	8·94	9·62
$\frac{1}{4}$	·36	·54	3	3·07	3·5	10	10·02	10·75
$\frac{3}{8}$	·49	·67	$3\frac{1}{2}$	3·55	4	11	11	11·75
$\frac{1}{2}$	·62	·84	4	4·03	4·5	12	12	12·75
$\frac{3}{4}$	·82	1·05	$4\frac{1}{2}$	4·51	5	13	13·25	14
1	1·05	1·31	5	5·04	5·56	14	14·25	15
$1\frac{1}{4}$	1·38	1·66	6	6·06	6·62	15	15·43	16
$1\frac{1}{2}$	1·61	1·90	7	7·02	7·62	16	16·4	17
2	2·07	2·37	8	7·98	8·62	17	17·32	18

Mr. Geipel gives the following rules :—

d = diameter in inches.
L = length in feet.
p = loss of pressure due to friction.
D = weight of steam in pounds per cubic foot.
Q = pounds of steam per hour.
v = velocity of flow in feet per minute.

$$Q = 3000\sqrt{\frac{p\,d^5\,D}{L}}$$

$$p = \frac{1}{9000{\cdot}000} \times \frac{Q^2\,L}{d^5\,D}$$

$v = 9170\sqrt{\dfrac{p\,d}{L\,D}}$, whence the Tables IV., V. are deduced.

TABLE III. OF EQUATION OF PIPES.

STANDARD STEAM AND GAS PIPES.

Dia.	½	¾	1	1½	2	2½	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17
½		2·27	4·88	15·8	31·7	52·9	96·9	205	377	620	918	1,292	1,767	2,488	3,014	3,786	4,904	5,927	7,321	8,535	9,717
¾	2·60		2·05	6·97	14·0	23·3	42·5	90·4	166	273	405	569	779	1,096	1,328	1,668	2,161	2,615	3,226	3,761	4,282
1	7·55	2·90		3·45	6·82	11·4	20·9	44·1	81·1	133	198	278	380	536	649	815	1,070	1,263	1,576	1,837	2,092
1½	24·2	9·30	3·20		1·26	3·34	6·13	13·0	23·8	39·2	58·1	81·7	112	157	190	239	310	375	463	539	614
2	54·8	21·0	7·25	2·26		1·67	3·06	6·47	11·9	19·6	29·0	40·8	55·8	78·5	95·1	119	155	187	231	269	307
2½	102	39·4	13·6	4·23	1·87		1·83	3·87	7·12	11·7	17·4	24·4	33·4	47·0	56·9	71·5	92·6	112	138	161	184
3	170	65·4	22·6	7·03	3·11	1·66		2·12	3·89	6·39	9·48	13·3	20·9	23·7	31·2	39·1	50·6	61·1	75·5	88·0	100
4	376	144	49·8	15·5	6·87	3·67	2·21		1·84	3·02	4·48	6·30	8·61	12·1	14·7	18·5	23·9	28·9	35·7	41·6	47·4
5	686	263	90·9	28·3	12·5	6·70	4·03	1·83		1·65	2·44	3·43	4·69	6·60	8·00	10·0	13·0	15·7	19·4	22·6	25·8
6	1,116	429	148	46·0	20·4	10·9	6·56	2·97	1·63		1·48	2·09	2·85	4·02	4·86	6·11	7·91	9·56	11·8	13·8	15·6
7	1,707	656	226	70·5	31·2	16·6	10·0	4·54	2·49	1·51		1·41	1·93	2·71	3·28	4·12	5·34	6·45	7·97	9·31	10·6
8	2,435	936	322	101	44·5	23·8	14·3	6·48	3·54	2·18	1·43		1·35	1·93	2·33	2·92	3·79	4·57	5·67	6·60	7·52
9	3,335	1,281	440	137	60·8	32·5	19·5	8·85	4·85	2·98	1·95	1·37		1·41	1·71	2·14	2·77	3·35	4·14	4·83	5·50
10	4,393	1,688	582	181	80·4	42·9	25·8	11·7	6·40	3·93	2·57	1·80	1·32		1·21	1·52	1·97	2·38	2·94	3·43	3·91
11	5,642	2,168	747	233	103	55·1	33·1	15·0	8·22	5·05	3·31	2·32	1·70	1·28		1·26	1·63	1·88	2·43	2·83	3·22
12	7,087	2,723	938	293	129	69·2	41·6	18·8	10·3	6·34	4·15	2·91	2·13	1·61	1·26		1·30	1·57	1·93	2·26	2·58
13	8,657	3,326	1,146	358	158	84·5	50·7	23·0	12·6	7·75	5·07	3·56	2·60	1·98	1·53	1·22		1·21	1·49	1·74	1·98
14	10,600	4,070	1,403	438	193	103	62·2	28·2	15·4	9·48	6·21	4·35	3·18	2·41	1·88	1·50	1·22		1·24	1·44	1·64
15	12,824	4,927	1,698	530	234	125	75·3	34·1	18·7	11·5	7·52	5·27	3·85	2·92	2·27	1·81	1·48	1·21		1·17	1·35
16	14,978	5,758	1,984	619	274	146	88·0	39·9	21·8	13·4	8·78	6·15	4·51	3·41	2·66	2·12	1·73	1·42	1·18		1·14
17	17,537	6,738	2,322	724	320	171	103	46·6	25·6	15·7	10·3	7·20	5·27	3·99	3·11	2·47	2·03	1·66	1·37	1·17	
18	20,327	7,810	2,691	840	371	198	119	54·1	29·6	18·2	11·9	8·35	6·11	4·63	3·60	2·87	2·35	1·92	1·59	1·36	1·16
20	26,676	10,249	3,532	1,102	487	260	157	70·9	38·9	23·9	15·6	10·9	8·02	6·07	4·73	3·76	3·08	2·52	2·08	1·78	1·52
24	42,624	16,376	5,644	1,761	778	416	250	113	62·1	38·2	25·0	17·5	12·8	9·70	7·55	6·01	4·92	4·02	3·32	2·84	2·43
30	75,453	28,990	9,990	3,117	1,378	736	443	201	110	67·6	44·2	31·0	22·7	17·2	13·4	10·7	8·72	7·14	5·88	5·03	4·30
36	120,100	46,143	15,902	4,961	2,193	1,172	705	319	175	108	70·4	49·3	36·1	27·3	21·3	16·9	13·9	11·3	9·37	8·01	6·85
42	177,724	68,282	23,531	7,341	3,245	1,734	1,044	473	259	159	104	73·0	53·4	40·5	31·5	25·1	20·5	16·8	13·9	11·9	10·1
48	249,351	95,818	33,020	10,301	4,554	2,434	1,465	663	363	223	146	102	75·0	56·8	44·2	35·2	28·8	23·5	19·4	16·6	14·2
Dia.	½	¾	1	1½	2	2½	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17

ACTUAL INTERNAL DIAMETERS.

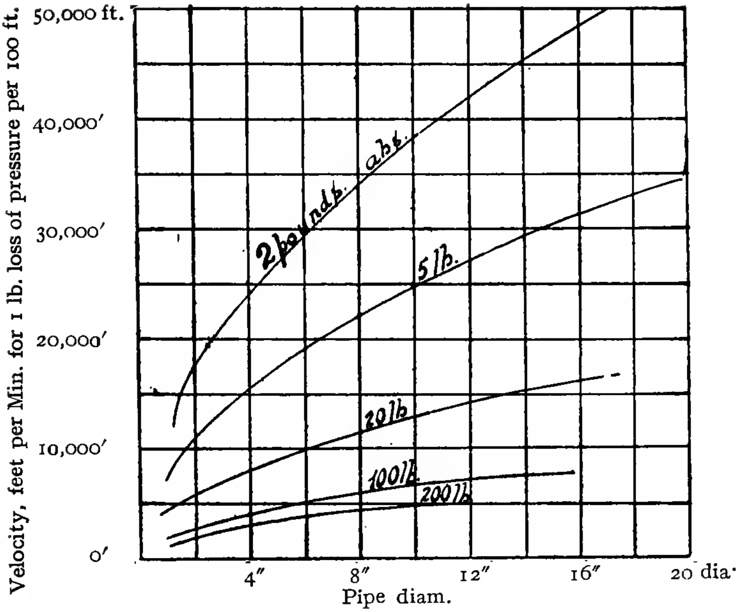
TABLE IV.

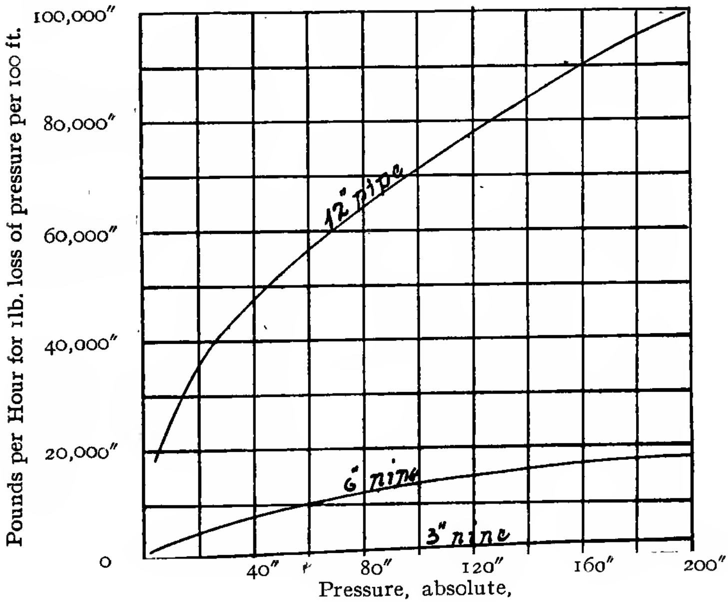
Diameter of Pipe. Inches.	Absolute Pressure. Pounds per Square Inch. Velocity in Ft. per Minute with 1 lb. Loss of Pressure per 100 Ft.				
	2	5	20	100	200
1	12000	7790	4070	1900	1380
2	17000	1100	5760	2690	1950
3	20850	13520	7070	3290	2390
6	29500	19100	9900	4660	3380
9	36100	23400	12050	5700	4140
12	41600	26900	14100	6580	4770
15	46600	30200	15600	7400	—
18	51100	33200	17100	—	—
24	58900	38150	—	—	—

TABLE V.

Diam. of Pipe. Inches.	Absolute Pressure. Pounds per Square Inch. Pounds of Steam per Hour with 1 lb. Drop of Pressure per 100 Ft.							
	2	5	10	20	50	100	150	200
1	22·8	35·2	48·7	67·6	104	144	174	200
2	129	199	275	382	590	815	984	1130
3	356	549	760	1054	1620	2245	2715	3120
6	2015	3100	4290	5960	9170	12700	15350	17640
9	5550	8550	11820	16400	25300	35000	42300	48600
12	11380	17500	24300	33700	51800	71700	86600	99600
15	19900	30610	42400	58900	90500	125500	—	—
18	31400	48400	67000	93000	143000	—	—	—
24	64400	99200	137000	190500	—	—	—	—

The accompanying diagrams are drawn from these formulæ.


Velocity, feet per Min. for 1 lb. loss of pressure per 100 ft.
50,000 ft.
40,000′
30,000′
20,000′
10,000′
0′
4″
8″
12″
16″
20 dia.
Pipe diam.
2 pounds abs.
5 lb.
20 lb
100 lb
200 lb


Pounds per Hour for 1lb. loss of pressure per 100 ft.
100,000″
80,000″
60,000″
40,000″
20,000″
0
40″
80″
120″
160″
200″
Pressure, absolute,
12″ pipe
6″ pipe
3″ pipe

For other losses of pressure, multiply the tabular numbers by the square root of the new loss. For other lengths of pipe $= L$ feet, multiply the tabular numbers by $\frac{10}{\sqrt{L}}$.

The values of $\sqrt{d^5}$ are best obtained by means of logarithms : they are given here up to 40 inches in Table VI.

TABLE VI.—$\sqrt{d^5}$.

1	1	11	401·3	21	2020·9
2	5·66	12	498·8	22	2270·1
3	15·6	13	609·3	23	2537·0
4	32·0	14	733·4	24	2821·8
5	55·9	15	871·4	30	4929·5
6	88·2	16	1024·0	40	10119·3
7	129·6	17	1191·6		
8	181·0	18	1374·6		
9	243·0	19	1573·6		
10	316·2	20	1788·9		

The length of pipe equal to a globe valve or to an opening into a pipe is given as $L_1 = 8{\cdot}66 \dfrac{d}{1 + \dfrac{3{\cdot}6}{d}}$

The length equivalent of an elbow is—

$$L_2 = 5{\cdot}76 \frac{d}{1 + \dfrac{3{\cdot}6}{d}}$$

For different diameters the equivalent lengths thus figure out in Table VII.

TABLE VII.

Diameter in Inches.	Equivalent Length to	
	Globe Valve or Pipe Opening.	Elbow.
1	1·9	1·3
2	6·2	4·2
3	7·9	5·2
6	32·5	21·6
9	55·6	37·0
12	79·9	53·2
15	100·5	69·6
18	129·9	85·9
24	180·6	123·8

Properly speaking, the opening to a pipe should be by a short converging piece, the wider end of which has an area about 10 per cent. greater than the pipe in order to allow for the *vena contracta* effect. Boiler mounting blocks do approximate to this shape, but their good effect is spoiled by the usually clumsy anti-priming pipe, which is not led up to the mouthpiece in an easy curve, and is usually plugged into the mouthpiece in such a way as to destroy the effect of this.

The resistance of openings and elbows by the

above rule is given in the Table VII. on page 21, and it will be observed that the results, in the smaller sizes, are much below the figures given on page 13.

CHAPTER III

Materials

STEAM PIPES are made of one of the four following materials :—

Cast Iron, Wrought Iron, Steel, Copper.

Cast Iron.

No material is so convenient or has been so largely employed as cast iron. Though a material of no flexibility, cast iron is strong and cheap, and with care can be cast sound and free from blemishes. The flanges are readily faced in the lathe, for which purpose stout bars carrying the centreings are commonly employed. Bolt holes are easily drilled. Pipes can be cast to any convenient length, and in brief, cast iron is without a serious rival for general purposes up to pressures of 100 pounds per square inch gauge pressure. Above that pressure the safety of cast iron admits of doubts ; above 120 pounds very serious doubts are to be entertained. The stresses in steam piping are not so much those of pressure as those which arise from expansion due to temperature changes and from water hammer, and, perhaps even more seriously, from forcing pipes to fill places which they do not fit properly. Some of these stresses are increased by pressure, viz., those due to expansive movements, and the high temperatures of superheat also have the same effect. Above 100 pounds, therefore, cast iron

should not be employed. True, junction pieces, valve bodies, etc., are still made by reputable firms, of cast iron up to 200 pounds pressure, and, while the author would condemn this practice, it is perhaps but fair to state that in such cases the choice of metal, the care in casting, and the rejection of faulty bodies, combine with the abnormal stoutness of parts to render such castings very much less unsafe than ordinary pipe castings from a jobbing foundry, with more or less uncertain coring and no special selection of the iron.

For exhaust pipes, however, cast iron holds the field. Exhaust pipes are usually larger, much larger than the steam pipe to the same engine, for it is their duty to carry the same steam in a wet condition and at much smaller pressures. To enable exhaust pipes to be tightly jointed the flanges require to be stout and to be faced. They should be cast from metal of good quality and tough, and not too hard to tool easily. Pipes should be cast vertically if they are to be reasonably safe against floating of the cores. The common fault of cast-iron pipes is the chaplet, which does not become melted fast in the pipe and causes blow holes, which admit air and vitiate the vacuum.

A usual rule for pipe thickness is $\frac{D.P.}{4000} = 0{\cdot}5$, when $D =$ diameter in inches, and $P =$ pressure, but this rule will give too small a thickness for exhaust pipes, and no pressure should be assumed less than, say, 4 pounds per square inch for each inch

of diameter of pipe. Flanges are made one-third thicker than the pipe body. Their duty is greater than the mere withstanding of pressure stress. In practice they are subject to very severe stresses of error which arise when pipes do not fit their places and joints are screwed up much too severely for good workmanship.

TABLE VIII.

THICKNESS OF CAST-IRON PIPES.

	in.	in.	in.	in.	in.	in.
Diameter . . .	4	5	6	7	8	9
Thickness. . .	$\frac{3}{8}$	$\frac{1}{16}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{9}{16}$
Diameter . . .	10	12	14	16	18	20
Thickness. . .	$\frac{5}{8}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{7}{8}$	$\frac{7}{8}$

Neither of the above rules is suited for a large range of diameters. A more adaptable rule is to make the thickness of the pipe $T = \frac{D + 4}{16}$. This rule serves for pipes from 2 to 12 inches, up to 100 pounds pressure.

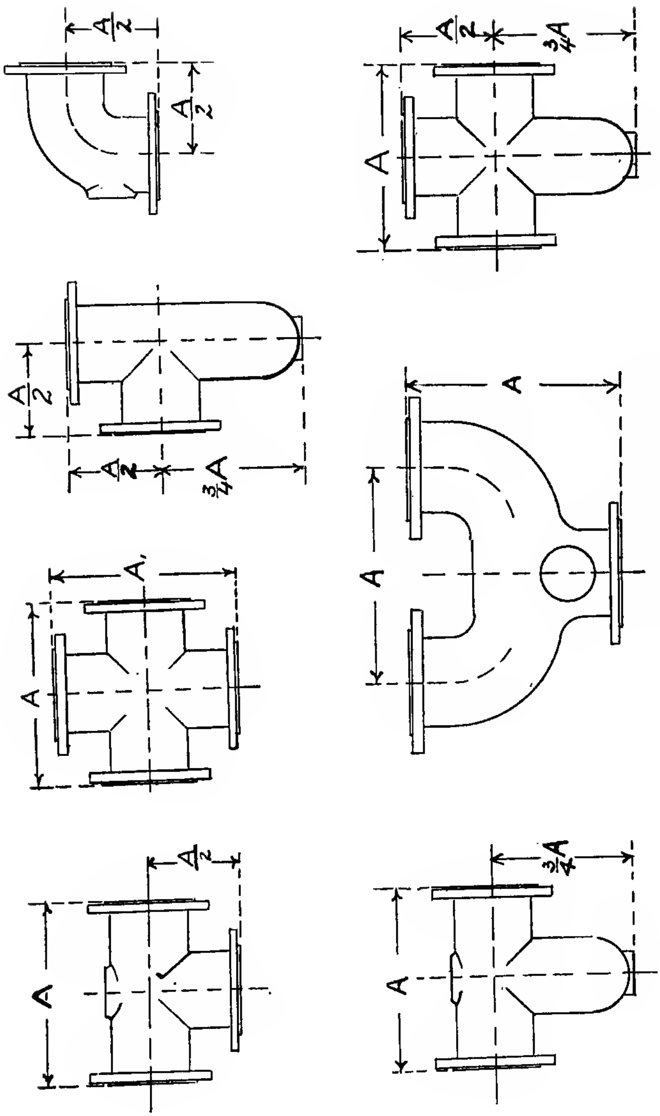
Above 100 pounds the rule is $T = \frac{D.P.}{4000} + \frac{1}{2}$, as given above, but these last rules give a pipe unnecessarily heavy for exhaust purposes, for which the author's rule is $T = \frac{4\,D^2}{4000} + 0{\cdot}5 - \frac{1}{2\,D}$, the result being taken only to the nearest sixteenth of an inch. Thus a 20-inch pipe has a thickness $\frac{1600}{4000} + 0{\cdot}5 - \frac{1}{40} = {\cdot}875''$. The nearest sixteenth to this is $\frac{7}{8}$ inch, and this rule gives close results to ordinary practice,

as per Table VIII., which is that given by the Babcock & Wilcox Co. for exhaust pipes.

Cast-iron pipes can be obtained in the form of tees, small radius bends and large radius bends. Straight pipes are made in 9-feet lengths, or, for sizes below 3-inch bore, in lengths of 6 feet. Making-up lengths are, of course, made to any 'templet' length (*see* Templet). Crosses, pockets, and Y-pieces are also made, and the Table XXIIA., later, and figures 1–7, herewith, give the sizes of such pieces as made by the Babcock Company. It will be noted that all pieces of the same main size must always have equal overall dimensions. Thus the projection of the branch on a 10-inch ⊥ will always be made 13 inches, whether the diameter of the branch piece be 2 inches, 7 inches, or 10 inches, and, similarly, the dimensions A A_1 of a cross will always be the same as the dimension A of a T, while of course the dimension B will always be half of A. Unless these precautions are taken, piping systems are liable to prove very inconvenient to put together.

Some engineers do not hesitate to employ cast-iron junction pieces for the highest pressures, taking care to relieve the cast pieces of the stresses of expansion, but the author does not recommend this. When so used they are made specially stout, while stout pieces are used for lower pressure and lighter castings for exhaust steam. There is always some risk of a light casting getting into a high pressure line.

Engineers differ as to the mode of facing flanges. A flange faced right across makes an excellent joint



FIGS. 1, 2, 3, 4, 5, 6, 7.—CAST JUNCTION PIECES.

with woodite for steam pipes, or with a plain ring for exhaust pipes. Some engineers recess their flanges in the form of a shallow spigot and socket, as shown in Fig. 50, by Yates & Thom. This certainly is a safeguard against blowing out the joint ring. Such pipes are often difficult to take down or to fit with a new ring. The Babcock Co. have a narrow facing strip only round the pipe, and make the joint with a light corrugated ring of brass or copper, all the bolt pressure being concentrated on the narrow face. *See* Figs. 11, 12.

In Table IX. are given the dimensions of the cast-iron exhaust tees of the standard of the British Electric Traction Co., kindly supplied me by Mr. A. J. Lawson, of that Company, and shown in Fig. 8.

The only remark that might be made on these is that bolts of $\frac{1}{2}$-inch diameter, as needed for the $\frac{5}{8}$ holes, are smaller than is perhaps desirable, nothing less than $\frac{5}{8}$th bolt diameter being very satisfactory in practice, though the smaller size was put in to be proportional to the rule of bolt numbers in multiples of four.

This Company face pipe flanges straight across, with no projecting ring and no recess. Flanges faced flat across are often scored with two or three circular grooves put in to the depth of $\frac{1}{32}$ or $\frac{1}{16}$ with a single V-point tool, with the object of better holding the joint rings. These grooves should only be employed where soft rings can be used. That is, they must not be used where joints are made with simple copper wire, for steam will leak at the joint where a wire may run across a groove.

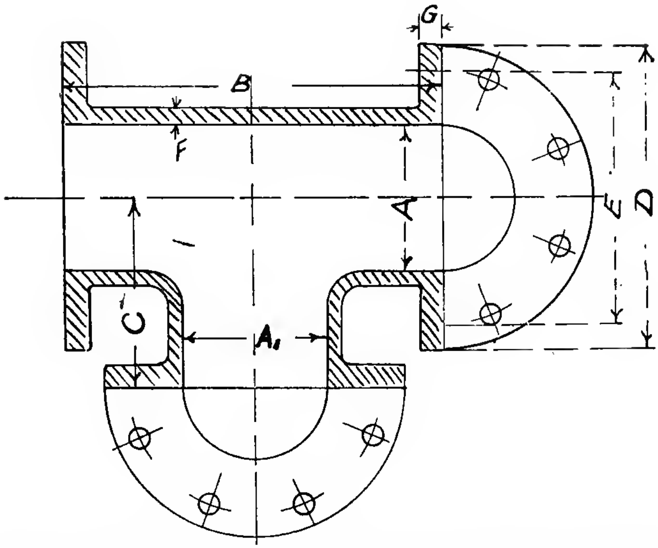


FIG. 8.—CAST-IRON EXHAUST TEES OF BRITISH ELECTRIC TRACTION CO.

TABLE IX.
Cast-Iron Exhaust Tees.

Size.	A	B	C	D	E	F	G	Bolt Holes.	
2½	2½	9	4½	7	5½	⅜	⅝	⅝	
3	3	9	5	7½	6	7/16	¾	¾	4
3½	3½	10	5	8	6½	7/16	¾	¾	
4	4	11	5½	9	7½	½	¾	⅝	
4½	4½	11	6	9½	8	½	¾	⅝	
5	5	13	6½	10½	8¾	½	¾	¾	
6	6	14	7½	12	10	½	¾	¾	8
7	7	15	8	13	10¾	9/16	⅞	¾	
8	8	16	8½	14	12	9/16	⅞	¾	
9	9	18	9	15	13	9/16	1	⅝	
10	10	19	10	16¼	14¼	⅝	1	¾	
11	11	20	10	17	15	⅝	1	¾	
12	12	22	11	19¼	17	¾	1⅛	⅞	12
13	13	23	12	20½	18	¾	1¼	⅞	
14	14	24	13	21½	19	¾	1¼	⅞	
16	16	27	14	24	21½	⅞	1⅜	1	

In case of reducing tees, no difference to be made in the dimensions *B* or *C*.

Bolts $\frac{1}{8}$ smaller than holes.

In Table X. and Fig. 9 the standards of the same Company are given for cast-iron steam tees, and in

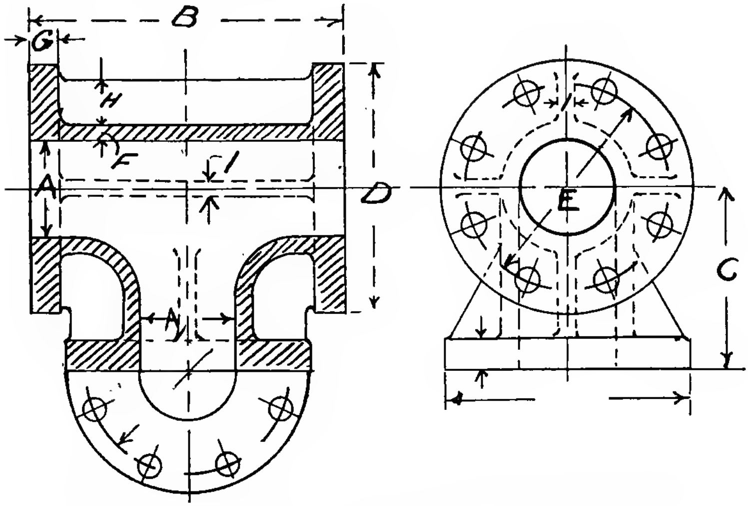


FIG. 9.—CAST-IRON STEAM TEES OF BRITISH ELECTRIC TRACTION CO.

TABLE X.

CAST-IRON STEAM TEES TESTED TO 200 LB.

Size.	A	B	C	D	E	F	G	H	I	Bolt Holes N	Bolt Holes S
3	3	$9\frac{1}{2}$	$5\frac{1}{2}$	$7\frac{1}{2}$	$6\frac{1}{8}$	$\frac{1}{2}$	$\frac{7}{8}$	$1\frac{1}{4}$	$\frac{3}{8}$	8	$\frac{5}{8}$
$3\frac{1}{4}$	$3\frac{1}{4}$	10	,,	$7\frac{3}{4}$	$6\frac{3}{8}$	,,	,,	,,	,,	,,	$\frac{3}{4}$
$3\frac{1}{2}$	$3\frac{1}{2}$	10	6	8	$6\frac{5}{8}$	,,	,,	,,	$\frac{7}{16}$	,,	,,
$3\frac{3}{4}$	$3\frac{3}{4}$	11	,,	$8\frac{1}{2}$	7	$\frac{9}{16}$	,,	,,	,,	,,	,,
4	4	$11\frac{1}{2}$	$6\frac{1}{2}$	9	$7\frac{1}{2}$	,,	1	$1\frac{1}{2}$	,,	,,	,,
5	5	13	$7\frac{1}{2}$	$10\frac{3}{4}$	$8\frac{7}{8}$	$\frac{5}{8}$	,,	$1\frac{3}{8}$	$\frac{1}{2}$	,,	$\frac{7}{8}$
6	6	15	8	12	10	$\frac{3}{4}$	$1\frac{1}{8}$	$1\frac{1}{2}$	,,	,,	1
7	7	16	$8\frac{1}{2}$	13	11	,,	$1\frac{1}{4}$	,,	,,	12	$\frac{7}{8}$
8	8	18	9	14	12	$\frac{7}{8}$	$1\frac{1}{4}$	$1\frac{5}{8}$	$\frac{5}{8}$	,,	$\frac{7}{8}$

Table XI. and Fig. 10 the same Company's standard for mild steel tees with standard flanges.

It will be noticed that these standards appear to

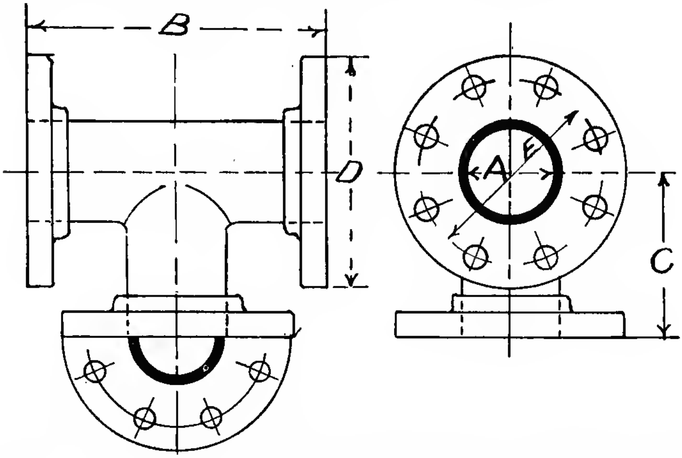


FIG. 10.—MILD STEEL STEAM TEES WITH STANDARD FLANGES OF BRITISH ELECTRIC TRACTION CO.

TABLE XI.

MILD STEEL STEAM TEES WITH STANDARD FLANGES.

Size.	A	B	C	D	E					N	S
$\frac{3}{4}$	$\frac{3}{4}$	$5\frac{1}{2}$	3	$4\frac{1}{8}$	$2\frac{7}{8}$					4	$\frac{5}{8}$
1	1	6	,,	$4\frac{1}{2}$	$3\frac{1}{4}$					,,	,,
$1\frac{1}{4}$	$1\frac{1}{4}$	7	$3\frac{1}{2}$	$4\frac{3}{4}$	$3\frac{1}{2}$					,,	,,
$1\frac{1}{2}$	$1\frac{1}{2}$	,,	4	$5\frac{1}{4}$	4					,,	$\frac{3}{4}$
$1\frac{3}{4}$	$1\frac{3}{4}$	8	$4\frac{1}{4}$	6	$4\frac{1}{2}$					,,	,,
2	2	,,	$4\frac{1}{2}$	$6\frac{1}{2}$	5					8	$\frac{5}{8}$
$2\frac{1}{4}$	$2\frac{1}{4}$	$8\frac{1}{2}$	,,	,,	,,					,,	,,
$2\frac{1}{2}$	$2\frac{1}{2}$	9	5	7	$5\frac{1}{2}$					,,	,,
$2\frac{3}{4}$	$2\frac{3}{4}$	$9\frac{1}{2}$	,,	$7\frac{1}{4}$	$5\frac{3}{4}$					,,	,,

have been designed somewhat on the idea of each size standing by itself of the best proportion according to the designer's views for the particular piece.

Thus the dimension C is not made one-half of B in every case, as it ought to be in the author's opinion. C is, however, always made the same for every tee of a given size A, no matter what the diameter of the branch or A_1.

In event of a + being required it would not measure equally over each rim of flanges, or if it did do so it would not work evenly with the tees of the same size. Though good in themselves and useful as a guide in flange proportion and generally, these standards should be changed in such respects in order to secure uniformity, so that the set of a T, the half-breadth of a +, and the set of a quarter-bend may all measure alike.

Mr. Venning advises that as regards bends in pipes which do not need to be proportioned to suit other junction pieces, the radius should not be less than five diameters of the pipe. An easy bend is better for the flow of steam. Small quarter-bends, however, when in particular situations, have to accommodate themselves to the size of other junction pieces, hence the dimension $\frac{1}{2}A$ in Fig. 4 corresponds with the dimensions of Figs. 1 and 2, which may be looked on as the leading junction pieces.

Copper.

As a material for steam pipes, copper has long held a place it can no longer claim with high temperature steam. Copper pipes are flexible because weak, and have been much used for lengths or bends intended to give way under stress.

Copper for pipes must not contain more than 0·7 of 1 per cent. of impurity, and the pipes should be solid drawn. A rule for brazed pipes is $\frac{D \times P}{10000} + 0{\cdot}125''$, where $D =$ pipe diameter in inches and $P =$ pressure by gauge per square inch.

Brazing must be looked on with great suspicion, though it may be employed in attaching flanges, which should be four times as thick as the pipes.

At a temperature of 360° F. the strength of copper is reduced 15 per cent. Copper is therefore to be employed cautiously for high pressures, and it is not a suitable material for conveying superheated steam.

The diminution of tenacity is shown in the annexed table.

DIMINUTION OF STRENGTH OF COPPER AT TEMPERATURES ABOVE 32°.

Temperature.	Loss of Tenacity. Per cent.	Temperature.	Loss of Tenacity. Per cent.
68°	2	638°	35
138°	5	748°	45
248°	10	788°	50
328°	15	838°	55
418°	20	938°	66
438°	22	968°	68
488°	25	1168°	88

The above figures must always be allowed for in calculating pipe strengths, the bursting pressure of which is found by the following rule:—

$\frac{t \times 2 \times s}{d}$ where t = thickness in inches.

s = tenacity in pounds per square inch.

d = pipe diameter in inches.

Good ordinary metal has the following values for s:—

Cast Iron	= 15,000.
Copper (cold)	= 30,000.
Wrought Iron	= 49,000.
Mild Steel	= 60,000.

In modern practice with superheated steam copper must not be assumed to have a tenacity above 16,500 pounds, and brazing at superheat temperatures becomes rotten.

In Admiralty practice copper pipes are wound with steel wire close laid as a precaution against ripping.

Mr. Ferranti, to avoid danger from large copper pipes, built up large pipes of a number of small pipes closely spaced in flange plates, which when bolted together gave a large number of pipes in cluster, but the system was expensive.

Solid-drawn copper pipes are said to be liable to longitudinal splits.

The ductility of copper is not great. When pulled apart by tension its reduction of area at fracture is small, and copper has lost any superior value it once possessed, perhaps very properly as compared with cast iron, in comparison with which copper first gained its character for elasticity and safety.

Mr. W. E. Storey states that a common cause of deterioration of copper is its contact when hot with reducing gases, such as coal gas, which makes the metal brittle. The same result is produced in the brazing hearth, when the air supply is insufficient. Such copper is properly to be termed gassed, rather than burnt, and this would tend to explain a fact and avoid a danger. He attributes failures in steam pipe to improper design, and urges solid-drawn tubes for bends with a radius at least three diameters of the pipe. He deprecates severe hydraulic tests. Though he is a maker of copper pipes, his advocacy is far from urgent, and engineers would be well advised to avoid copper for steam pipes, and especially superheated steam pipes, but copper may still be well employed for pipes containing water, such as feed pipes, the spring piece of a boiler blow-off—not on the boiler side of the blow-off cock, however.

Electro-deposited copper pipes are said to be 50 per cent. stronger than ordinary copper. The author has used such copper only in water work, and cannot speak to its use in steam work.

Flexible Metallic Pipes.

Flexible pipes are made by coiling into a closely interlocked helix a peculiarly folded strip of metal. This may be steel, zinc, brass, copper or a bronze alloy. These pipes are suitable even up to 300 pounds steam pressure. They are very flexible,

and are maintained steam tight by means of a packing of asbestos, which is the more tightly held in the folds of the helix the greater the pressure inside the pipe. Bronze is considered best for steam pipes, and these pipes are particularly suited for rapidly connecting boilers and engines on contractor's or temporary work. Flanges are attached by means of a screwed gland and collar, with asbestos packing, which holds firmly on the ridges of the pipe.

Any gap in a length of pipe can readily be made good by a piece of flexible pipe slightly long. It will accommodate itself to any flange angle. The author has no knowledge to go upon as to long continued durability, but there can be no doubt that it would form a perfect connection between a boiler and the main steam pipe.

If used where it is not supported at each end it might be advisable to restrain end movement, in order to keep the flange connections free from tension.

Where a bad foundation causes settlement and undue strains, a short length of flexible pipe would prevent all trouble It is well engineers should bear this flexible tubing in mind, for at times it may prove useful and of marked convenience.

Wrought Iron and Steel Pipes.

Pipes of steel up to 10 inches diameter can be had in weldless steel. Above that size, as well as below

TABLE XIa.

Wrought Steel Pipes, with Wrought Steel Flanges.

Internal diameter of Pipe in inches		$\frac{3}{4}$	1	$1\frac{1}{2}$	2	$2\frac{1}{2}$	3	$3\frac{1}{2}$	4
Thickness of Pipe in inches . .		9 W.G.	8 W.G.	6 W.G.	$\frac{3}{16}$	$\frac{3}{16}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$
Weight per foot of Pipe (approx.).	lb.	1·48	1·89	3·31	5·0	5·5	8·5	10·0	11·0
	kilos.	·673	·86	1·51	2·3	2·5	3·86	4·55	5·0
Weight per pair of Flanges (approx.).	lb.	2·8	5·8	6·9	17·0	20·0	25·0	26·0	38·0
	kilos.	1·27	2·63	3·09	7·73	9·10	11·35	11·8	17·3

Internal diameter of Pipe in inches		5	6	7	8	9	10	12	14
Thickness of Pipe in inches . .		$\frac{1}{4}$	$\frac{5}{16}$	$\frac{5}{16}$	$\frac{5}{16}$	$\frac{11}{32}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{7}{16}$
Weight per foot of Pipe (approx.).	lb.	14·0	21	24·5	27·5	35·5	41·0	49·0	67·0
	kilos.	6·35	7·5	11·1	12·5	16·1	18·6	22·3	30·4
Weight per pair of Flanges (approx.).	lb.	42·0	53·0	84·0	80·0	96·0	130·0	162·0	154·0
	kilos.	19·1	24·1	38·2	36·4	43·6	59·2	73·6	70·0

it, pipes can be had lapwelded. There is always some little doubt remaining as to the absolute soundness of a weld, and where such doubts are felt the riveted pipe may be relied on. The riveted pipe, when reasonable care has been taken in choosing good material, can be made to show a strength 70 per cent. of a solid pipe, and it can be relied on. Steel pipes and steel flanges are made of all forms, including straights, quarter-bends, quarter-bends with a length of straight, double quarter-bends joined by a bit of straight, and set-off or cranked lengths. Table XIA. (*see* previous page) will be useful in getting out approximate weights of pipe. It is copied from a list of the Babcock Co.

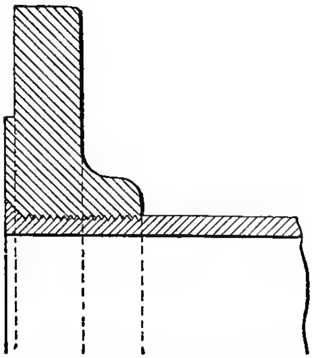
Steel for pipes should be of strictly mild quality, similar to boiler plate, with a tenacity of 24 to 27 tons per square inch and an elongation on tenacity test of not less than 20 per cent. in a length of eight inches.

The flanges of steel pipes are attached by three usual methods, viz., riveting, welding and screwing.

In the practice of the Babcock Co. pipes below 6 inches are screwed into their flanges (Fig. 11) and expanded by a tube expander. The Whitworth thread should be used. It has 11 threads per inch, in all sizes above $\frac{7}{8}''$. The coarser American thread does not produce such good work or so tight as the English or Whitworth thread. Pipes are faced and drilled after fixing the flanges.

Above 6 inches the Babcock Co. rivet on the

flanges (Fig. 12), and they frequently also rivet on branches, as in Fig. 13.



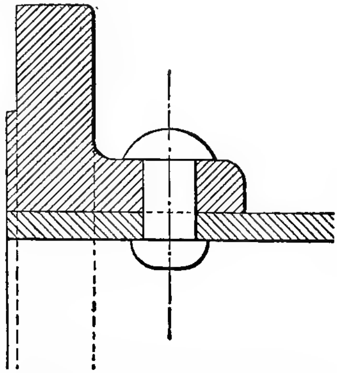


FIG. 11.—FLANGE SCREWED ON. FIG. 12.—FLANGE RIVETED ON.
BABCOCK & WILCOX CO.

The flanges of steel pipes are stamped out of solid forged pieces, and weigh as in the annexed Table 12 for high pressure pipes, Table 13 for

FIG. 13.—RIVETED BRANCHES. BABCOCK AND WILCOX CO.

lighter purposes, and Table 14 for heavy cast-iron flanges.

Riveted pipes are usually double-riveted lap-jointed scarfed down and tucked into the flange,

as in boiler-making practice; the longitudinal seams are thinned at the overlap and tucked into the ring seams.

The Mannesmann Co. of London make solid rolled steel tubes up to 12-inch external diameter, from 0·104 inch thick in 2-inch tubes up to 0·312 for sizes above $10\frac{1}{2}$. These pipes can be had with flanges

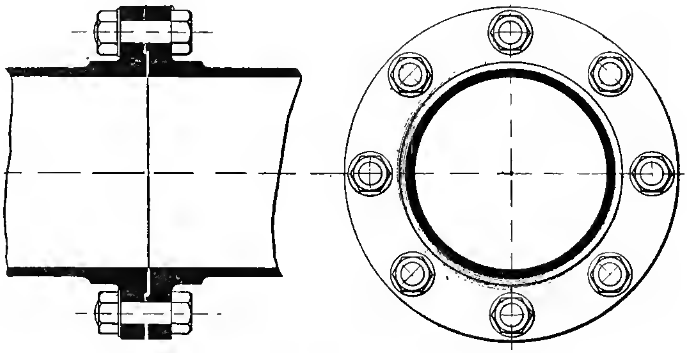


FIG. 14.—SOLID WELDED FLANGE (YATES & THOM).

attached by any approved method. They also manufacture pipes, the flanges of which are loose and are slipped on to the pipes, which are afterwards flanged or lipped up in various ways. These flanges, or lips, are drawn together by bolts through the loose heavy flanges and are offered for use with high pressures and superheated steam.

Steel pipes are also made with their flanges welded solid with the pipe, as shown at Fig. 14.

The steel pipe is *par excellence* the proper pipe for high pressure and for superheated steam. The tenacity of steel at 60,000 pounds per square inch

gives steel pipes of ordinary thickness a very large margin of strength, a 6-inch pipe at 200 pounds pressure, and $\frac{1}{4}$ inch thick, only carrying a unit stress of 2,400 pounds per square inch, or a 25-fold margin when solid drawn, and perhaps 15 to 20-fold if lap welded. The best practice for junction pieces is to make these of wrought steel, but this becomes

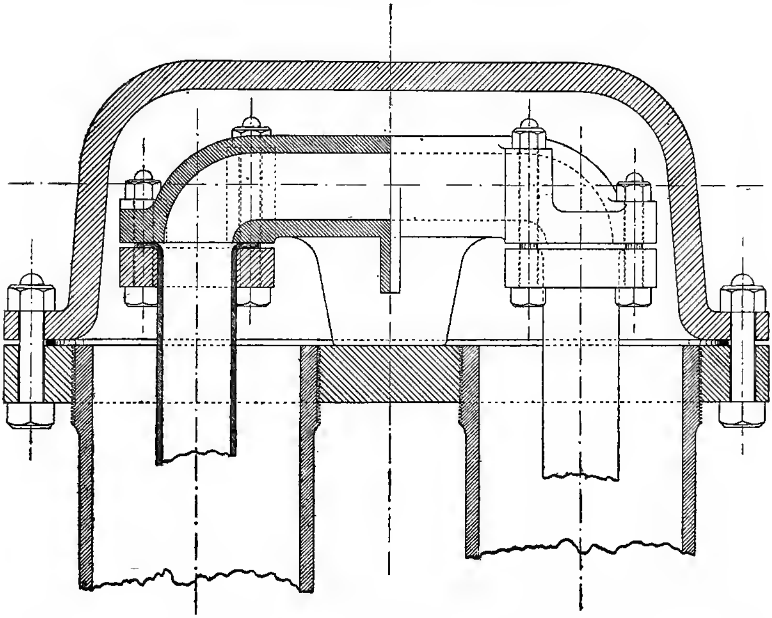


FIG. 15.—HEADER OF CRUSE CONTROLLABLE SUPERHEATER, SHOWING METHOD OF MAKING JOINTS.

expensive in the large sizes, and they are often made of cast steel, of similar pattern to cast iron, but they need not be so stout. The medium dimensions of cast-iron pieces should be ample for cast steel. Some engineers consider it quite good practice to employ cast-iron junction pieces, which, if stout

and carefully cast of strong metal, they look on as safe for the highest pressures. Perhaps they are safe when due provision is made to relieve expansion stresses, but their chief danger is perhaps from the sudden shock of water hammer. For superheated steam steel pipes are most desirable, especially in the superheater itself. As an example of the highest class of pipe work the steel pipes of the Cruse Controllable Superheater (Fig. 15) may be cited. These pipes are usually 6 inches external diameter, and $\frac{5}{16}$-inch thick. They are of solid rolled weldless steel. Their extremities are staved or thickened up for threading, so that the diameter at the base of the thread is a little in excess of the external diameter of the pipe body. The staved portion is threaded, and the pipes are simultaneously screwed at both ends into headers of $1\frac{1}{2}$-inch rolled steel plate. They are then expanded. The cover box which encloses the ends of two pipes with the coupling box of the internal 2-inch water-control pipe is of pressed mild steel, and the joint between cover and header plate is made by means of a solid ring of $\frac{3}{16}$-inch round copper wire. These joints have never been known to fail, and similar joints may be made between ordinary faced flanges by means of copper wire. The writer has made them with $\frac{3}{16}$-inch wire only looped into a circle and the ends simply crossed over each other, the bolts tightening the wire sufficiently to flatten the crossing to a steam-tight condition.

TABLE XII.

High Pressure Steam Flanges, Pattern A.

Internal Diameter of Pipe. Inches.	Outside Diameter of Pipe. Inches.	Diameter of Flange. Inches.	Approximate Weight.	
			Lb.	Kilos.
2	$2\frac{3}{8}$	7	$7\frac{3}{4}$	3·49
$2\frac{1}{2}$	$2\frac{7}{8}$	$7\frac{3}{4}$	9	4
3	$3\frac{1}{2}$	$8\frac{3}{4}$	$12\frac{1}{4}$	5·5
$3\frac{1}{2}$	4	9	$12\frac{3}{4}$	5·74
4	$4\frac{1}{2}$	10	$16\frac{1}{2}$	7·43
5	$5\frac{1}{2}$	11	22	9·9
6	$6\frac{5}{8}$	12	25	11·25
7	$7\frac{5}{8}$	14	43	19·35
8	$8\frac{5}{8}$	14	$35\frac{1}{2}$	16
9	$9\frac{11}{16}$	15	$46\frac{3}{4}$	21
10	$10\frac{3}{4}$	17	58	26
12	$12\frac{3}{4}$	$19\frac{1}{2}$	$72\frac{3}{4}$	32·74
14	$14\frac{7}{8}$	$21\frac{1}{2}$	$87\frac{1}{2}$	39·38

TABLE XIII.

Light Weight Steel Flange, Pattern B.

Internal diameter of Pipe. Inches.	Outside Diameter of Pipe. Inches.	Diameter of Flange. Inches.	Approximate Weight.	
			Lb.	Kilos.
$\frac{3}{4}$	$1\frac{1}{16}$	$3\frac{1}{2}$	1	·45
1	$1\frac{5}{16}$	$4\frac{1}{2}$	$2\frac{1}{2}$	1·13
$1\frac{1}{4}$	$1\frac{5}{8}$	$4\frac{1}{2}$	$2\frac{1}{4}$	1·02
$1\frac{1}{2}$	$1\frac{7}{8}$	5	$2\frac{3}{4}$	1·24
2	$2\frac{3}{8}$	6	$4\frac{3}{4}$	2·14
$2\frac{1}{2}$	$2\frac{7}{8}$	7	$6\frac{1}{4}$	2·81
3	$3\frac{1}{2}$	$8\frac{1}{4}$	$9\frac{1}{2}$	4·28
4	$4\frac{1}{2}$	$9\frac{1}{2}$	$14\frac{3}{4}$	6·64

HEAVY PATTERN CAST-IRON FLANGES, FACED AND DRILLED, FOR AUXILIARY, STEAM, FEED AND BLOW-OFF PIPES.

TABLE XIV.

Cast-Iron Flange.

Internal diameter of Pipe. Inches.	Outside Diameter of Pipe. Inches.	Diameter of Flange. Inches.	Approximate Weight.	
			Lb.	Kilos.
$\frac{3}{4}$	$1\frac{5}{16}$	$3\frac{1}{2}$	2	·9
1	$1\frac{1}{16}$	$4\frac{1}{2}$	$3\frac{1}{2}$	1·6
$1\frac{1}{4}$	$1\frac{5}{8}$	$4\frac{1}{2}$	$3\frac{1}{4}$	1·5
$1\frac{1}{2}$	$1\frac{7}{8}$	5	$3\frac{1}{4}$	1·5
2	$2\frac{3}{8}$	6	$6\frac{3}{4}$	3·1
$2\frac{1}{2}$	$2\frac{7}{8}$	7	$8\frac{1}{2}$	·4
3	$3\frac{1}{2}$	$8\frac{1}{4}$	14	6·4
4	$4\frac{1}{2}$	$9\frac{1}{2}$	18	8·2

Steel Alloy.

Some boiler makers will provide what they call steel pipes, which are really malleable iron, or so-called steel alloy. They are often tough, but are difficult to face, and are apt to suffer from bad spots, which leak out steam. No doubt such pipes are much superior to cast iron, but they are not equal to steel pipes with either screwed or welded flanges.

Though Tables XII., XIII., XIV. are given as the Babcock standards, it may be doubted if it is worth while having flanges of different diameters for auxiliary pipes, etc.

At Fig. 16 is shown the riveted flange as made by Yates & Thom, with the recessed face and projecting shallow spigot, which, while it is

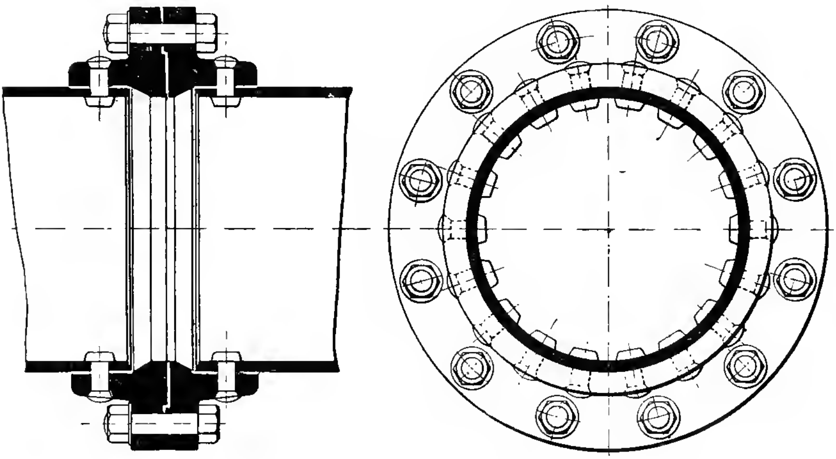


FIG. 16.—RIVETED FLANGE (YATES & THOM).

apt to make it difficult to pull pipes apart, is very efficacious in preventing joint rings from blowing out. Pipes put together with indiarubber rings in these recesses are sometimes very difficult to part. Plain flanges with sheet joints can be sawn apart with an old saw, which will clean off the flange faces ready for a new sheet to be inserted.

Socketed Joints.

A form of joint of extreme neatness not much employed is the socketed joint. Sockets are usually screwed upon the pipe as tightly as possible, coming to a stop where the thread dies away into the barrel. The next length of pipe is screwed upon the previously erected length, and it is obvious that the last length must be flanged at one end, for it cannot be screwed two ways at once. But the flange is not always even possible, and socketed pipes are joined finally by means of a "long thread" or "connector," which consists of a piece of pipe of any length, one end of which is threaded a long way so that the threaded portion will hold the full length of a socket as well as a back nut. In making a final joint by this long thread the socket is fully screwed back, the ends of the pipes to be joined are brought together, and the socket is then screwed forward upon the end to be joined up. For each thread it advances on one pipe it leaves the other, and of course is somewhat slack upon the long thread. The back nut is then screwed against the socket, a thread of asbestos with cement being wrapped round the thread and forced tightly against the socket end by the back nut.

There should be a back nut at each end of the socket for steam work.

Artesian pipes are faced off to dead lengths of 10 feet, or other desired lengths, and have special sockets which screw exactly half length on each

pipe and tighten up on the last of the thread just as the faced-off ends meet at the middle of the socket. Such pipes could be made to form a steam-tight joint by means of a ring of copper between the ends. It is possible that socketed pipes will come more into use for special work, as they are very neat, offer the minimum of surface for loss of heat, and can be covered with sectional or other covering to look very neat, having no flanges. The long thread affords every facility for making up lengths exactly. It is, however, certain that socketed pipes are troublesome to take apart. The sockets become very fast. They are best put together with Dixon's smear-grease, a compound of mineral oil and graphite, which is said not to become hard.

As regards wrought-iron pipes these are lap-welded for steam purposes, are to be treated as described for steel, and it is the author's belief are often supplied of steel to fill wrought iron orders.

Flexible Metallic Tubing.

For temporary work, flexible metallic tubing coiled up from pecular doubled or folded steel strip, interlocked and flexibly packed with a thread of asbestos or other fibrous matter, may be employed. It may be obtained attached to flanges, and for rapid connection is easily put in to occupy the place of making-up lengths not yet arrived from the makers.

It is best made of bronze for steam purposes, as to which a further note is made under the head of "Copper."

American pipe of 1½ and up to 2-inch sizes is screwed 11½ threads per inch. Above that size it is screwed 8 threads, which seems coarse to English engineers, and is not so good as our Whitworth 11 threads for light work.

The following is the American pipe list abridged for lap-welded wrought-iron pipe:—

Inside Diameter.	Outside Diameter.	Weight per Foot.	Threads per Inch.
1½	1·900	2·68	11½
2	2·375	3·60	11½
2½	2·875	5·73	8
3	3·500	7·54	—
3½	4·000	9·00	—
4	4·500	10·66	—
4½	5·000	12·34	—
5	5·563	14·50	—
6	6·625	18·76	—
7	7·625	23·27	—
8	8·625	28·18	—
9	9·625	33·70	—
10	10·750	40·06	—
11	12·000	45·95	—
12	12·750	49·00	—
13	14·000	54·00	—
14	15·000	58·00	—
—	16·000	61·77	—
—	18·000	70·00	—
—	20·000	77·57	—
—	22·000	85·47	—
—	24·000	93·37	—

American pipes are screwed with a taper per inch of length of screw of $\frac{1}{32}$ up to 8 inches diameter and $\frac{1}{64}$ above that size.

As with English pipes, the inside diameter of American W.I. or steel pipe is not the nominal diameter, but varies as the thickness of the pipes varies, the outside diameter being constant for any nominal size.

WHITWORTH PIPE THREADS.

Internal Diameter.	External Diameter.	Diameter at bottom of thread.	Threads per Inch.
$\frac{1}{2}$	·826	·734	14
$\frac{3}{4}$	1·04	·949	14
1	1·309	1·192	11
$1\frac{1}{4}$	1·650	1·533	11
$1\frac{1}{2}$	1·882	1·765	11
2	2·347	2·23	11
$2\frac{1}{2}$	3·00	2·882	11
3	3·485	3·368	11
4	4·340	4·223	11

Pipes are not made exactly to their nominal inside diameters. All pipes, whatever their strength, have equal outside diameters for the same nominal internal diameter. Any change of thickness adds to or subtracts from the inside dimensions. The outside diameter of English and American pipes differ very slightly.

Messrs. John Spencer, Ltd., say that for general

work there is nothing to beat lapwelded steel pipes, with solid welded flanges, and branches riveted on, for sizes from 2-in. bore to 12-in. inclusive. For larger pipes riveted flanges are preferred, and for low pressure and small pipes, screwed flanges. This firm's list of standard flange diameters, drilling, etc., and also thickness of pipes for both high and low pressure steam main work is annexed. The thickness of pipes is given for straights; bends are always made somewhat thicker :—

TO 120 LB. PRESSURE.

Bore.	Diameter.	Thickness.	No. of Holes.	Diameter of Pitch Circle.	Size of Bolts.
$\frac{1}{2}$ in.	$3\frac{1}{2}$ in.	$\frac{1}{2}$ in.	4	$2\frac{1}{2}$ in.	$\frac{7}{16}$ in.
$\frac{3}{4}$,,	$3\frac{3}{4}$,,	$\frac{1}{2}$,,	4	$2\frac{3}{4}$,,	$\frac{7}{16}$,,
1 ,,	$4\frac{1}{2}$,,	$\frac{1}{2}$,,	4	$3\frac{3}{8}$,,	$\frac{1}{2}$,,
$1\frac{1}{4}$,,	5 ,,	$\frac{1}{2}$,,	4	$3\frac{5}{8}$,,	$\frac{1}{2}$,,
$1\frac{1}{2}$,,	$5\frac{1}{2}$,,	$\frac{1}{2}$,,	4	4 ,,	$\frac{1}{2}$,,
2 ,,	6 ,,	$\frac{5}{8}$,,	4	$4\frac{5}{8}$,,	$\frac{1}{2}$,,
2 ,,	$6\frac{1}{2}$,,	$\frac{5}{8}$,,	4	5 ,,	$\frac{1}{2}$,,
$2\frac{1}{4}$,,	$6\frac{1}{2}$,,	$\frac{5}{8}$,,	4	5 ,,	$\frac{1}{2}$,,
$2\frac{1}{2}$,,	7 ,,	$\frac{5}{8}$,,	6	$5\frac{1}{2}$,,	$\frac{1}{2}$,,
3 ,,	8 ,,	$\frac{3}{4}$,,	6	$6\frac{1}{4}$,,	$\frac{5}{8}$,,
$3\frac{1}{2}$,,	$8\frac{1}{2}$,,	$\frac{3}{4}$,,	6	$6\frac{3}{4}$,,	$\frac{3}{4}$,,
4 ,,	9 ,,	$\frac{3}{4}$,,	6	$7\frac{1}{4}$,,	$\frac{3}{4}$,,
5 ,,	$10\frac{1}{2}$,,	$\frac{7}{8}$,,	6	$8\frac{3}{4}$,,	$\frac{3}{4}$,,
6 ,,	12 ,,	1 ,,	8	10 ,,	$\frac{3}{4}$,,
7 ,,	$13\frac{1}{2}$,,	1 ,,	8	$11\frac{1}{2}$,,	$\frac{3}{4}$,,
8 ,,	15 ,,	$1\frac{1}{8}$,,	8	$12\frac{3}{4}$,,	$\frac{3}{4}$,,
9 ,,	16 ,,	$1\frac{1}{8}$,,	10	$13\frac{3}{4}$,,	$\frac{3}{4}$,,
10 ,,	17 ,,	$1\frac{1}{8}$,,	10	$14\frac{3}{4}$,,	$\frac{7}{8}$,,
11 ,,	18 ,,	$1\frac{1}{4}$,,	12	$15\frac{3}{4}$,,	$\frac{7}{8}$,,
12 ,,	19 ,,	$1\frac{3}{8}$,,	12	$16\frac{3}{4}$,,	$\frac{7}{8}$,,

TO 200 LB. PRESSURE.

Bore.	Thickness of Pipe.	Thickness of Flange.	No. of Holes.	Diameter of Pitch Circle.	Size of Bolts.
$\frac{1}{2}$ in.	10 g.	$\frac{1}{2}$ in.	4	$2\frac{1}{2}$ in.	$\frac{7}{16}$ in.
$\frac{3}{4}$,,	9 ,,	$\frac{1}{2}$,,	4	$2\frac{3}{4}$,,	,,
1 ,,	8 ,,	,,	,,	$3\frac{3}{8}$,,	$\frac{1}{2}$,,
$1\frac{1}{4}$,,	7 ,,	,,	,,	$3\frac{5}{8}$,,	,,
$1\frac{1}{2}$,,	6 ,,	,,	,,	4 ,,	,,
2 ,,	,,	$\frac{5}{8}$,,	,,	$4\frac{5}{8}$,,	$\frac{5}{8}$,,
2 ,,	,,	$\frac{5}{8}$,,	,,	5 ,,	,,
$2\frac{1}{4}$,,	5 ,,	,,	,,	,,	,,
$2\frac{1}{2}$,,	$\frac{1}{4}$ in.	,,	6	$5\frac{1}{2}$,,	,,
3 ,,	,,	$\frac{3}{4}$,,	,,	$6\frac{1}{4}$,,	$\frac{3}{4}$,,
$3\frac{1}{2}$,,	,,	,,	,,	$6\frac{3}{4}$,,	,,
4 ,,	,,	,,	,,	$7\frac{1}{4}$,,	,,
5 ,,	,,	$\frac{7}{8}$,,	8	$8\frac{3}{4}$,,	,,
6 ,,	,,	1 ,,	,,	10 ,,	,,
7 ,,	,,	,,	,,	$11\frac{1}{2}$,,	,,
8 ,,	$\frac{5}{16}$,,	$1\frac{1}{8}$,,	,,	$12\frac{3}{4}$,,	$\frac{7}{8}$,,
9 ,,	,,	,,	10	$13\frac{3}{4}$,,	,,
10 ,,	,,	,,	,,	$14\frac{3}{4}$,,	,,
11 ,,	$\frac{3}{8}$,,	$1\frac{1}{4}$,,	12	$15\frac{3}{4}$,,	,,
12 ,,	,,	$1\frac{3}{8}$,,	12	$16\frac{3}{4}$,,	,,

Flange diameters as on previous list.

For making steam joints, Taylor's corrugated brass rings give the best result, and they also strongly advise the adoption of a facing strip $\frac{1}{32}$-in. deep on the flanges of all high pressure pipes.

The following empirical formula is given for getting out quickly and accurately the length of tube in a right-angle bend ; it is found to give excellent results :—Take the sum of the two arms and deduct 5 inches for every foot of radius, plus $\frac{3}{4}$ inch for

stretching. Thus, for example: a bend setting 6 feet with a radius of 3 feet—

$$6' \, 0'' + 6' \, 0'' = 12', \text{ less } 1' \, 3\tfrac{3}{4}'' = 10' \, 8\tfrac{1}{4}''.$$

The lengths of tube in the pipe will be 10′ $8\frac{1}{4}''$.

In getting out steam mains, the chief point to watch is free expansion, which is obtained by using a sufficient number of lapwelded steel bends of large radii, or by the insertion of special expansion pieces. The rate of expansion of steam pipes is taken by Messrs. Spencer as follows:—

Copper.	*Steel.*	*C. Iron.*	*W. Iron.*
·012 in.	·00822 in.	·0077 in.	·0082 in.

per 10 feet for a rise of 10 degrees Fahr.

The number of joints is to be minimized by using as long pipes as possible. Appended is a list giving what may be considered stock lengths for various sizes of pipes.

Short bends and cast steel elbows should not, of course, be used when they can be avoided, nor should cast iron be used for bends at all; in fact one should always endeavour to eliminate it from steam main work, especially where there is high pressure and superheated steam, as cast iron deteriorates very much when carrying superheated steam.

Two other very important points are the proper supporting and draining of ranges; if the former is not well done vibration will ensue, and if the latter is insufficient water hammer is set up; excessive

vibration will cause leaky joints, and may lead to very serious consequences.

The following formulæ for thickness of pipes are given :—

t = thickness of pipe in inches.
p = pressure per square inch.
d = diameter (internal) in inches.

For lapwelded wrought iron, $t = \dfrac{p\,d}{6000}$

,, ,, cast-iron, $t = \dfrac{p\,d}{3500} + \frac{1}{4}$.

,, Copper (brazed) $t = \dfrac{p\,d}{6000} + \frac{1}{16}$.

,, ,, (solid drawn) $t = \dfrac{p\,d}{6000} + \frac{1}{32}$.

These are as used by the Board of Trade, and hold good, generally speaking, for bores up to 12-in., and water Pipes of 200 lb. per sq. in. Another very fair formulæ for cast-iron up to 200 lb. per sq. in. water-power, is $t = \dfrac{p \times d}{4000} + \frac{1}{2}$; this gives somewhat heavier castings.

A good formulæ for thickness of flanges on cast-iron pipes is :—

$T = 1{\cdot}4 \times t \times {\cdot}15$,

where T = thickness of flange,
t = ,, ,, pipe.

For bolts :—

$d = {\cdot}83t + {\cdot}3$ d = diam. of bolt,
$n = {\cdot}6D + 2$ n = number ,,
t = thickness of pipe.
D = diameter ,,

The following bursting pressure of steel pipes is given :—

Diameter, Pipes in Inches.	6 in.	7 in.	8 in.	9 in.	10 in.	11 in.	12 in.
Thickness.							
$\frac{1}{8}$ in.	1600	1372	1200	1066	960	873	800
$\frac{3}{16}$,,	2400	2058	1800	1599	1440	1309	1200
$\frac{1}{4}$,,	3200	2744	2400	2132	1920	1745	1600
$\frac{5}{16}$,,	4000	3430	3000	2665	2400	2181	2000
$\frac{3}{8}$,,	4800	4116	3600	3198	2880	2617	2400
$\frac{7}{16}$,,	5600	4802	4200	3731	3360	3053	2800
$\frac{1}{2}$,,	6400	5488	4800	4264	3040	3489	3200
	13 in.	14 in.	15 in.	16 in.	17 in.	18 in.	19 in.
$\frac{1}{8}$ in.	738	685	640	600	565	533	505
$\frac{3}{16}$,,	1107	1029	980	900	847	800	757
$\frac{1}{4}$,,	1476	1372	1280	1200	1129	1066	1009
$\frac{5}{16}$,,	1845	1715	1600	1500	1411	1333	1261
$\frac{3}{8}$,,	2214	2058	1920	1800	1693	1600	1513
$\frac{7}{16}$,,	2583	2401	2240	2100	1975	1866	1765
$\frac{1}{2}$,,	2952	2744	2560	2400	2257	2133	2017

Stock Lengths, 16 to 17 ft. up to 10″ diam.
,, ,, 15 to 16 ,, ,, ,, 11″ to 12″ diam.

CHAPTER IV

Expansion

THE necessity of cooling-off boilers for cleaning and repair, and the fact that some boilers are spare and cold, causes the connecting pipes of the boilers to the main to vary in length, not only as between their hot and cold conditions, but as between one boiler and others. In a length of 100 feet a steam pipe may expand 2 inches or more. The coefficient of expansion of cast iron is 0·00000618 per degree Fahrenheit =0·0000111 per 1° C. Wrought iron expands 0·00000656 per 1° F. = 0·0000118 per 1° C.

Between the temperature at which the pipes were fixed, say, 59° F. = 15° C., and the temperature of steam at 190 pounds gauge pressure per square inch, say, 383° F. = 195° C., the expansion will be about 4½ inches per 100 feet.

Mr. Venning says a good practical rule is to allow 1 inch for each 50 feet. Beyond the superheater the expansion effects will be even greater, for the temperature may be 653° F. = 345° C., a rise of nearly 600° F. = 330° C., or nearly 6½ inches per 100 feet above the cold length, for at high temperatures the expansion coefficients become greater (*see* table, p. 68).

Obviously, therefore, such variations must be provided for. In the case of a long battery of boilers with a straight main steam pipe athwart them, if the middle of the pipe was anchored fast the two ends would extend considerably. This would push outwards the branch pipes of the boilers to an amount gradually increasing as the distance from the central point was increased.

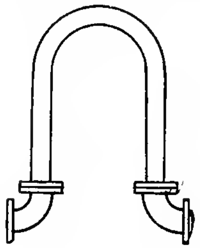


FIG. 17.

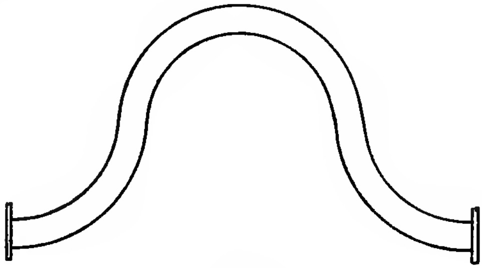


FIG. 18.

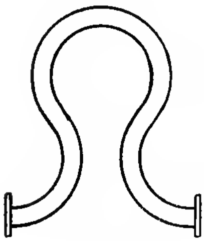


FIG. 19.

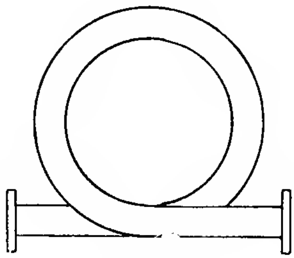


FIG. 20.

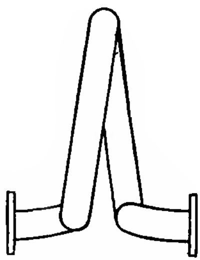


FIG. 21.

FORMS OF EXPANSION BENDS.

The branch pipes would give way by their elasticity, but they would of course exert a twisting stress upon the mounting block, the vertical pipe above this, and the valve, if this was at the top of the vertical branch. In long lengths of main, therefore, bends of the form of Figs. 17 to 21 are employed.

If circumstances are such that one of these expansion lengths is to be fixed, its place should be as nearly central as possible in the main, which would be anchored, if anchored at all, at one-fourth its length from each end, thus dividing it up into four distinct lengths, and limiting expansion in any one section to one-fourth the total. The anchoring of a pipe in this way prevents the steam pressure on the extreme ends of the pipe from acting to pull open the expansion bend, but if anchored in such a way as to prevent lateral movement there would be introduced a stress in the nearest boiler branch pipes. Anchoring, therefore, should be longitudinal only. In arranging for expansion bands the loop should, if possible, be horizontal, so as to obviate water pockets. If vertical, there must be a small connecting pipe looped down from the straight main to carry water across the gap of horizontal continuity. The expansion bend cannot be allowed to hang downwards from the pipe unless the bottom of the loop is drained by a trap, and this position must not be used, if possible to be avoided.

Figs. 17, 18, 19 are usual types of bends. Fig. 20 is convenient where there is a change of level greater than the pipe diameter, for the lateral displacement of the loop may be little or considerable.

In Fig. 15 the stress, says Mr. Stromeyer, is torsional.[1]

[1] *The Manchester Steam Users' Association.* Memorandum by Chief Engineer, June, 1901.

A bend is more elastic than a straight of equal length from crown to crown. Thus, if in Fig. 17 the two straight arms were connected by a rigid casting in place of by a bend, this form would be the most rigid of all the arrangements shown. Mr. Stromeyer represents the elasticity of the two straight arms by 2. Then each of the forms, Figs. 18, 19, 20, will spring an amount $2 \times 6 = 12$. Fig. 17, consisting of two-thirds straight pipe and one-third bend, will be represented by $2\frac{2}{3}$. In Fig. 21 the elasticity is 21.

The permissible stretch of any form varies with the mean height of the loop, is inversely as the square of the diameter and independent of the thickness (in all practical thicknesses).

The force to stretch a bend is proportional, however, to the thickness and to the diameter squared. Bends are therefore made thin and weak if they have to relieve stress on a weak point, such as a cast-iron valve or junction piece, but for expansion of long pipes the bends may be of the same material and thickness as the pipes.

Copper, once so much used for bends, is not so very suitable, though it may be made thin. Its elastic limit is low, and it has less spring than mild steel or wrought iron. It is a metal that grows brittle with age, and it is dangerous at high temperatures.

With bends of 4 feet crown to crown, and a diameter of pipe of 6 inches, Mr. Stromeyer gives the following (Table XV.) of safe extensions of bends :—

TABLE XV.

Material.	Two Straight Pipes.	Fig. 17.	Figs. 18, 19.	Fig. 20.	Fig. 21.
Steel .	0·21	0·42	0·74	0·74	2·60
Copper .	0·12	0·23	0·41	0·41	1·45

The values in the table, except for Fig. 21, may be doubled where the pipes they relieve have freedom for lateral movement, and, again, this double value may be again doubled if the bends are initially stretched by the same amount they will be compressed when hot, so that a copper bend of Figs. 18, 19, 20 type would, if erected cold and stretched 0·82, allow of a total difference of length between cold and hot of 1·64 inches, or enough for a length of 50 feet of main.

In a battery of four boilers, as very commonly arranged in cotton mill work, *see* Fig. 22, as arranged by Yates & Thom. The branch pipes of the boilers are about 12 feet long from crown of steam mounting block to crown of main steampipes. The only relief necessary here is given by the two bends and long straight piece of pipe between the boiler main and the engine, the starting valve of which is not in line with the boiler main. A similar double bend connects the high-pressure cylinder with the first initial-pressure cylinder.

Expansion joints are sometimes made of the form

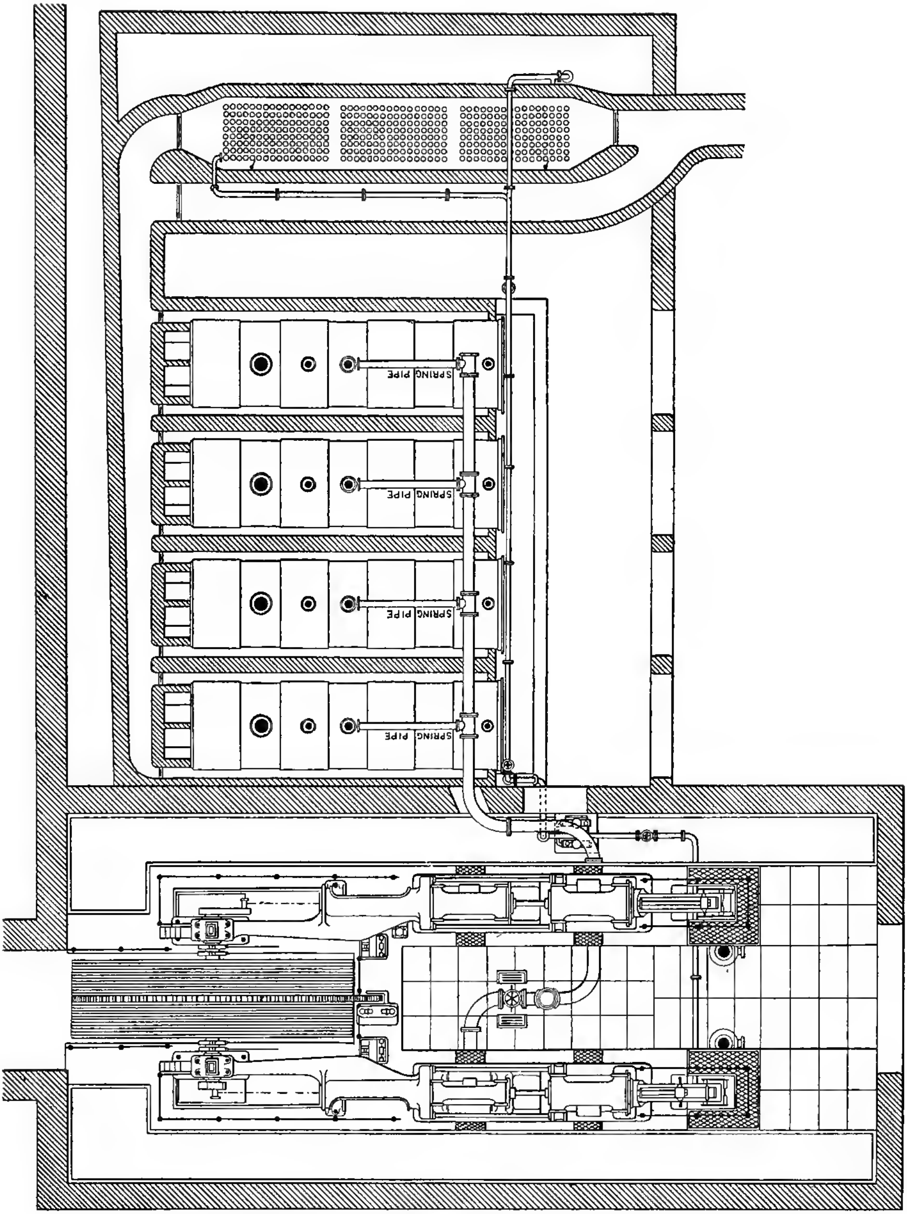
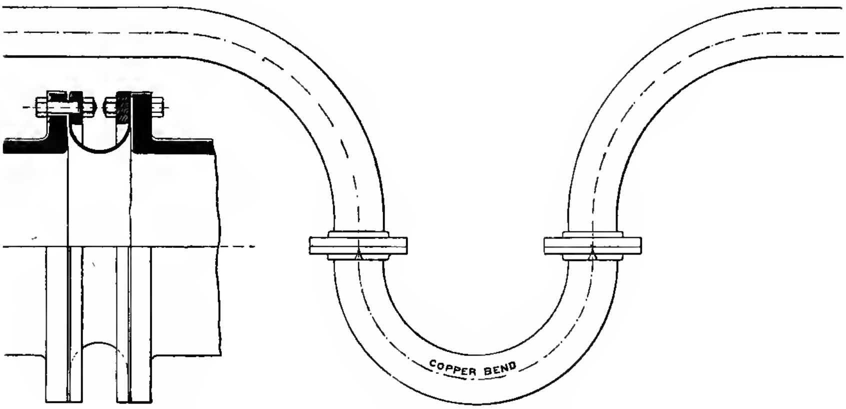


FIG. 22.—GENERAL ARRANGEMENT OF COTTON MILL-PLANT (YATES & THOM).

of Fig. 23. These are small and compact, but are apt to become choked with deposit and to split. Larger expansion discs are made by riveting together two flatly dished plates. These are also liable to choke with deposit, and they also offer a large surface to the steam pressure, which exerts



FIGS. 23, 24.—EXPANSION JOINTS (YATES & THOM).

a very heavy thrust and helps to nullify the movement of pipes when these are wanted to contract. They resist the very movement they are designed to accommodate, and they are therefore only to be recommended for exhaust pipes, in which case the outside pressure exceeding the inside pressure tends to move the pipes in the same direction as the expansion will move them.

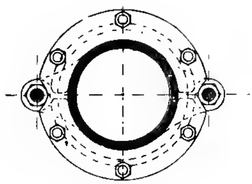


FIG. 25—END VIEW.

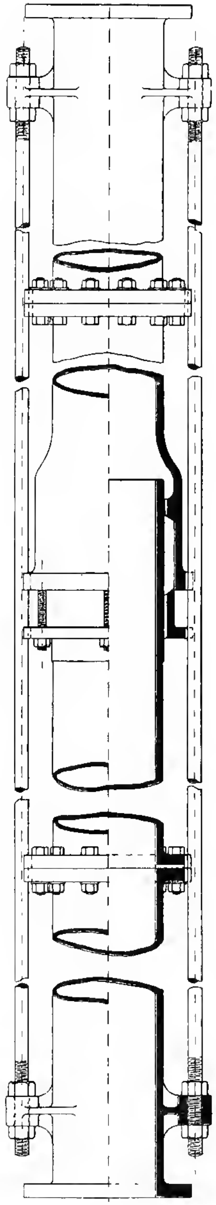


FIG. 25.—TELESCOPIC EXPANSION JOINT (YATES & THOM)

Expansion joints of the form of Fig. 25 are sometimes used. This particular one, as made by Messrs. Yates & Thom, consists of a sliding pipe and stuffing box to provide movement on each side of the joint the pipe has legs upon it which are joined across the joint by two long bolts that must be strong enough to resist the steam pressure on the area of the pipe. In a length of pipe served by such a joint as this the bolts would be of a length

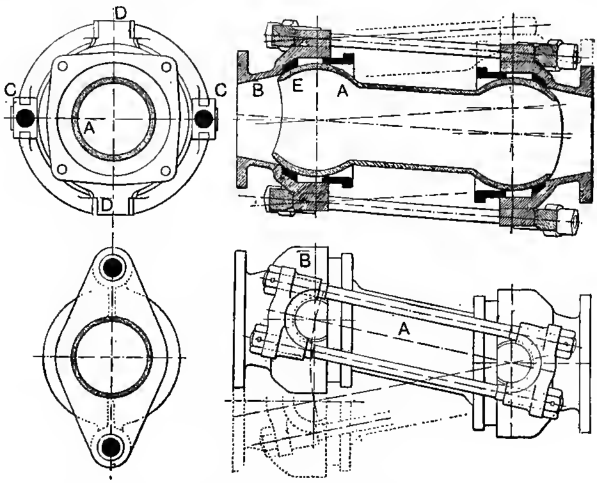


FIG. 26.—HARTER'S SWIVELLING EXPANSION COUPLING.

equal to half the length of the pipe. This class of expansion joint is only advised where spring bends cannot conveniently be introduced. Shorter expansion joints are made complete in overall lengths of from 27 to 32 inches, according to size, i.e. about $20 + 1{\cdot}5\,d$ inches, and weighing from 75 to 80 pounds per inch of pipe diameter, according to size, the larger ones being of the heavier proportion.

Harter's joint is shown in Fig. 26. This explains

itself. The joint provides a universal movement of a pipe in the way of bending, but this joint is not an expansion joint to be placed in the length of a pipe. It is rather to be placed in the branch pipes of each boiler, so as to allow free movement of the main pipe without straining of the branch pipes. Where economy is specially desirable these joints would not be necessary on the first (or perhaps also the second) boiler, on each side of the middle point of the main.

If this joint is so arranged that when cold it lies in a straight line, or nearly so, with the boiler branch pipe, it is obvious that when the main steam pipe lengthens, these swivelling joints will be displaced laterally, and will be no longer straight and in line. The flanges they connect will therefore need to approximate each other by the amount of the versed sine of the arc of swivelling. As, however, the boiler branch pipe becomes longer also by heat expansion, this angular movement of the swivel piece will provide to some extent for this expansion. Thus, in a swivelling length of 30 inches a movement of 1 inch in the main steam pipe would imply an angle of 2 degrees, the versed sine of which is ·0006 or 0·018 inch. This is only about one-eighth of what a branch pipe of 5 feet in length would expand at usual pressures. But if the swivel piece be already placed at an angle of 4 degrees, when cold, its movement only 2 degrees further would increase the versed sine movement by ·0048, or eight-fold. Obviously, therefore, a boiler branch

of a length of 5 feet, with a 30-inch swivel-piece placed when cold 2 inches out of line, would allow for a movement of the steam main of 1 inch when hot, and would take up the expansion of the boiler branch. If boilers and engines are carefully fixed to drawn positions, as they may be, and pipes are made to the same measures so as to come right without final make-up lengths, as is also not merely possible but practicable, then it would be possible so to arrange the whole scheme of pipes as, by placing the swivel at greater initial angles towards the end of a range of boilers, to eliminate all stresses of expansion. Even if such stresses were reduced to half or a third, it would be a desirable thing to accomplish.

In many power stations the boiler branch pipes enter the main steam pipe, and the engine branch pipes are taken out from points very near to them. Where possible, expansion is provided by bending the boiler branches so as to enter the steam main at the top. If the engine pipes leave the main from its upper side also, the main acts as a water separator, and must be drained. If the engine branches leave from the bottom of the pipe, all water must then be dealt with at the engine separators. Both the boiler and engine branches may enter at the opposite sides of the main without bends. In this case the engine branches are usually bent down to the engine some feet further on, and the separator is placed in the horizontal part of the engine branch. With water-tube boilers, where there is a clear gangway

behind the boiler seating, the boiler branches have been brought down by bends to the steam main placed in the gangway, and the engine branches have been carried up from the main, bent through the engine-room wall, thence carried a few feet, and bent down to the engine stop-valve.

This arrangement is very elastic, because the various vertical pipes are several feet long. The main must be carried above the passage ways between pairs of boilers, at least 6 feet above floor. It must also be drained.

In case of very long mains the expansion, if not otherwise provided for, may be allowed for by an expansion T-piece. The one pipe has a closed end and passes right through the head of the T, being made steam-tight by glands at each end. That part of the pipe inside the T is perforated by slots to permit steam to pass to the T and to the pipe, which is rigidly bolted to the single end of the T. The expansion of the long pipe can take the place of sliding, and there is no end-thrust.

The disadvantage is, that the steam has to take a sharp square bend and pressure is lost. Substantially the provision of suitable bends and sufficiently long branches is alone necessary for general work, and a scheme of pipe work must be carefully thought out so that expansion shall not be concentrated at one point, but shall be well distributed throughout the system, bearing in mind always that any one boiler may be at rest between two working boilers, or *vice versa* ; and due consideration

must be given to each movement that will occur under the extremes of conditions.

The expansion of any other bends than those of 4 feet × 6 inches given in the table, page 59, can be calculated by the rule given, or $E = \frac{e\,H}{4} \times \frac{36}{d^2}$, or $E = \frac{9\,H.\,e.}{d^2}$, where d is the diameter of pipe in inches, H[1] the loop height in feet, and e is the tabular expansion for 6-inch pipe, E being the permissible expansion of the pipe sought. Then E, as found, may be doubled or quadrupled, according as the pipe is free for lateral movement, or is extended when cold.

The calculation of the expansion of any length of pipe is made by the following formulæ :—$S = L\,t\,f$, where S is the movement sought, L is the length in feet, t the range of temperature over which the expansion occurs, and f is the coefficient of expansion per foot of pipe per degree of temperature. In a long main taking branches from several boilers, the middle of the main may, as stated, be anchored fast. This is easily effected when the pipe is carried on a bracket, as a cap may be bolted over the pipe, but not so as to prevent lateral movement. At other points the pipe may be suspended by rods from brackets above, and a short spring may be placed between the bracket and the nut of the suspender.

Anchoring of one point is desirable for the purpose of checking the vibration which is often set

up by the connection of the pipes to the engines, or by the intermittent impulses of the moving steam. A pipe which swings in this way may usually be steadied without locking it fast, if a stop is placed against a point of chief movement to limit the amplitude of the vibration and destroy the natural rhythm of the movement.

Expansion Coefficients.

The expansion of cast iron between 32° and 42°, or a range of 180° F., is given in Molesworth as 0·0011, wrought iron, 0·0012, and copper, 0·0018.

Kempe gives the coefficient of linear expansion per degree Fahr. as follows :—

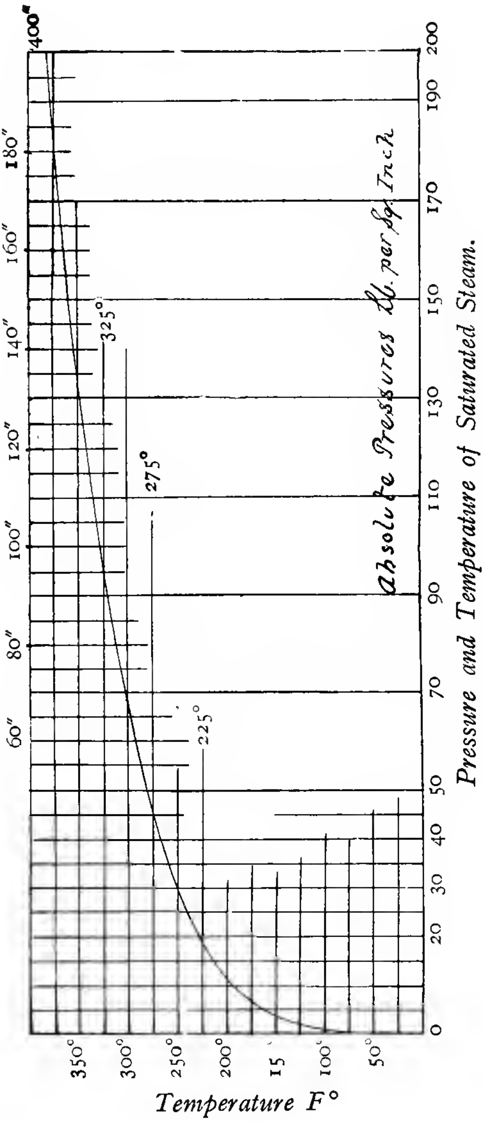
Metal.	Coefficient.	Tested Between.
Cast Iron . .	0·00000618	32°–212°
Steel . . .	0·00000600	32°–212°
Wrought Iron .	0·00000895	32°–572°
,, ,, .	0·00000656	32°–212°
Copper . .	0·00000955	32°–212°
,, . .	0·00001092	32°–572°
Firebrick . .	0·00000275	32°–212°
Good Red Brick	0·00000305	32°–212°

From this it would appear that at temperatures of superheated steam the coefficient of expansion per degree Fahrenheit for steam pipes may be taken as 0·000008, which is nearly 2 inches in 50 feet for a temperature rise of 400° F.

While undoubtedly stresses are often very severe and manifest themselves by failures of cast-iron

junction pieces, and by weeping joints and even rivets, it must not be necessarily inferred that all the movement calculated does actually occur to produce stress. Much may be done in the way of giving counter-stresses initially, and cold to reduce the working stresses when hot. Nor can we assume that the boiler does not move from its cold position. The expansion of firebrick is about half that of iron, and under boiler conditions it is much hotter and its actual expansion is as much or more. The seating of a Lancashire boiler lengthens as much as the boiler, the movement one way of the steam outlet block on a boiler may be as much as the expansion the other way of the steam pipe. An *ultra* precision and refinement is therefore not called for, but it is easy to see that the expansion of the boiler branch pipe may be very fairly balanced by compelling the boiler steam-drum to expand from a determined and anchored point. Practice has taught what may and what may not be done, but special cases may require special consideration, and for these the methods indicated may be followed. The use of superheated steam not only increases the pipe temperatures but also increases the coefficient of expansion, which (as per table p. 68) becomes 0·000009 nearly between cold and 572° F. Such an expansion as this figure implies is very considerable. At the same time, probably, the pipes and bends are more yielding and take up the stresses by further movement. Where superheaters are placed behind the boilers, as in case of

Lancashire type boilers, the pipes to the superheater have only the ordinary expansion. But



the superheater is connected with the main, and this is to be considered in the design.

The annexed diagram will be of use in ascertaining the temperature of saturated steam from the known pressure. With the prospect of superheat being added, the temperatures found by the diagram should be increased by about 150° F. when calculating probable expansions to be provided for.

CHAPTER V

Strength of Pipes

THE thickness of a steel or wrought-iron pipe necessary for screwing is more than sufficient for all ordinary sizes at high pressures.

The stress on the material of a pipe per inch of length is the product of the diameter and the pressure per square inch. This product, divided by twice the thickness of the pipe, gives the unit stress.

Good wrought iron may be assumed to have an ultimate strength of 20 tons per square inch, and steel of 28 tons. On this basis, with a marginal factor of 5, the stress permissible will be about 9,000 pounds for iron and 15,000 pounds for steel. Double riveted joints have a 70 per cent. efficiency, and if lapwelding be allowed to have the same, the unit working tenacity will be 6,300 pounds and 10,500 pounds respectively.

Thus a 12-inch pipe at 200 pounds, if only $\frac{1}{4}$-inch thick, only carries a unit stress of 4,800 pounds. For very large work, even to 24-inch pipes, the stress on pipes $\frac{1}{4}$-inch thick would only be 9,600 pounds at 200 pounds pressure, or within the working stress of mild steel. Steam pipes of ordinary manufac-

turer's thickness of tube walls are thus of ample and excessive strength when only double riveted. Bad welds should be provided against by hydraulic test up to 12,000 pounds in iron and 21,000 pounds in steel, as calculated on the actual thickness, which will usually exceed $\frac{1}{4}$-inch in pipes of even 10 inches diameter.

Very large pipes may be worth riveting with butt strips. Solid rolled pipes can be calculated to stand a unit stress of 15,000 pounds, and need not exceed the thickness proper to this stress so long as the threading at the flanges does not unduly reduce them and render them liable to crack off at the flange. Probably steam pipes are made too heavy, and being so they throw undue stresses on cast-iron junction pieces, which are therefore made unduly clumsy to stand the stresses.

Solid rolled pipes with flanges double-riveted appear to offer the maximum strength per unit of weight. Flanges screwed on to large thin pipes involve the weakening due to the threading.

The Whitworth thread is the best for pipe threads, as it is finer than the American thread and cuts less of the pipe away. Its main dimensions are given here within Table XVI., up to 4 inches. All larger sizes have the same thread of 11 threads per inch. Pipes of all kinds are of the same diameter outside. The nominal inside diameter becomes less as the pipe is made stronger. Threads may thus be all standard, but some pipe makers do not make to the Whitworth standard even to-day.

TABLE XVI.

WHITWORTH PIPE THREADS.

Size.	No. of Threads per Inch.	External Diam.	Diameter at Bottom of Thread.
$\frac{1}{2}$	14	0·8257	0·7342
$\frac{3}{4}$	14	1·041	0·9495
1	11	1·309	1·1925
$1\frac{1}{4}$	11	1·650	1·5335
$1\frac{1}{2}$	11	1·8825	1·765
2	11	2·347	2·2305
$2\frac{1}{2}$	11	3·0013	2·8848
3	11	3·485	3·3685
$3\frac{1}{2}$	11	3·912	3·7955
4	11	4·339	4·223

According to the Board of Trade, the strength of copper pipe, well made, with brazed joints, is, working pressure $= \frac{6000\ (T - \frac{1}{16})}{D}$, where T and D are the thickness and diameter in inches. If solid drawn and not over 8 inches diameter, the $\frac{1}{16}$-inch is replaced by $\frac{1}{32}$-inch.

Wrought iron lapwelded pipes of good material are allowed $\frac{6000 \times T}{D}$ = working pressure.

It is well when pipes are screwed into their flanges to taper the hole on the face side of the flange and roll over the pipe end. This provides an extra longitudinal strength. Pipes too thin in relation to their diameters, if exposed to heavy end pull, will jump clean out of their sockets without showing any injury to threads. They cannot do this so well

with rigid flanges, but too much reliance must not be placed on mere screwing.

Rivets, again, must be proportioned for shear to stand the maximum possible end stress in the pipes, which is not likely to be greater than the steam pressure multiplied by the "area of pipe." This "area of pipe" may be greater than the nominal area, for it is the area enclosed within the joint ring. Rivets may be allowed a working stress in shear of 10,000 pounds per square inch, and will always be found to have a large excess over the stress on even the largest pipes. Rivets, therefore, are proportioned for steam tightness.

Longitudinal rivet seams, of course, are proportioned as in boiler work.

CHAPTER VI

Anti-Priming Pipes and Outlet Valves

IT was formerly the practice to lead off the steam pipe from a boiler by way of a steam dome. But these being found not to abolish priming have been discarded, and the steam pipe is attached directly to a mounting block, which, as elsewhere stated, ought to taper, so as to allow steam to enter it without loss by *vena contracta* effect.

Inside the boiler is fixed the anti-priming pipe. This is a length of pipe which usually extends each way from the steam outlet block, into which it is fitted by a short neck The sides and top of the pipe are slotted with holes, the joint area of which should be 25 per cent. greater than the area of the steam pipe supplied. Each branch of the anti-priming pipe should have a diameter of about three-fourths at least of the steam pipe. The anti-priming pipe is held to place by hangers attached to riveted lugs on the boiler crown.

Illustrations for ordinary practice are given in Figs. 27 and 28. It would be good practice to enlarge the central part of pipe so that it could be brought down at the bottom, and an easy leading

curve made to the outlet, so as not to oppose too sudden a bend at this point.

The holes in the pipe should be confined to the upper quarter or third of the circumference, and it is usual to drill a drain hole at the lowest point to let out any water. There ought properly to be a drain pipe carried to below water level in order to free the drainage water from the rush of steam.

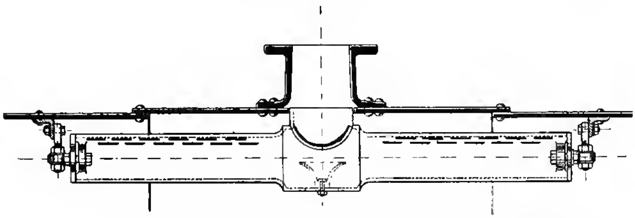


FIG. 27.—ANTI-PRIMING PIPE (YATES & THOM).

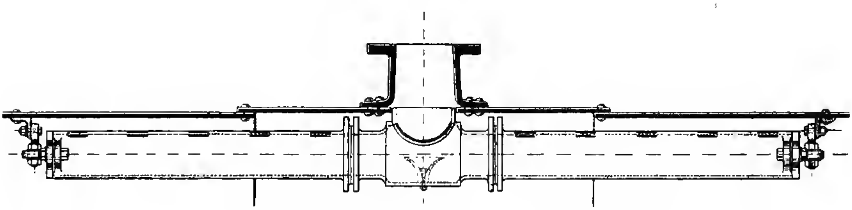


FIG. 28.—ANTI-PRIMING PIPE (YATES & THOM).

Anti-priming pipes are made about 6 feet long. Their object, of course, is to collect steam from a considerable length of a boiler, so as to avoid local rushes and formation of vortex whirls which would pick up water. They are sometimes made of copper of great length. The author has seen them extended to a double or ring main of over 20 feet of thin copper, perforated or slit. This seems needlessly long, for a sufficient spread of the intake can

usually be obtained in a length of 6 feet, and the area of holes can be got in to the requisite extent of 1·25 times the steam-pipe area. If this is exceeded, the steam will enter over too limited a length of the anti-priming pipe and defeat the object of the pipe. It is probable that excellent anti-priming pipes could be made from slitted brass sheet similar to the slit brass used for covering driven wells, but experiment is wanting to determine the amount of steam that will pass by a given length of slit. In the water sheets the slits are merely cuts without removal of material, and are very effective to keep back sand, while passing water much more freely than it could pass small holes of many times the area of opening.

The Steam Outlet Valve.

These have always been of the mushroom variety, and have necessarily been opened with or against the pressure in the boiler.

When the valve opens against the pressure it can of course be easily shut, and the pressure keeps it shut. It possesses the fatal objection that when shut it depends on the strength of its spindle to withstand the steam pressure in the main steam pipe, where other boilers are at work. This is a fatal objection, because it endangers the safety of the boiler cleaner or inspector in the idle boiler. The shut-down type has the objection that in order to prevent it being opened when the boiler is at rest it must be loose on its spindle, so that it only opens

by steam pressure, and these loose valves float on the outgoing steam and keep up a constant ringing sound, which, however, is objectionable only as showing wear.

The author would prefer the full-way double-faced slide-valve type to either of the other two forms, though this variety has the fault that it can be opened during the presence in the boiler of work-men, and ought to be specially safeguarded by a lock.

The position of the outlet valve is important. If when placed close to the boiler the steam pipe extends vertically above the valve for any distance before bending away to the main, or as in some cases the main is immediately on the top of the vertical pipe, then the vertical length of pipe becomes a water pocket should the boiler be at rest. This necessitates the employment of a drain pipe taken off the valve body at the lowest point and fitted with an automatic steam trap, for it is too dangerous to trust to the opening of a drain cock, when perhaps a labourer is sent to open up the valve. The water in the vertical pipe would all be blown forward and produce a most violent concussion at the first square end, or at the blank end of the main, or at any interposed resistance. Drain pipes and steam traps are a source of loss at any time. Good practice eliminates both if possible. To do this at the stop valve the vertical pipe is placed directly on the boiler block, and the stop valve, if of mushroom type, is made an angle valve,

and placed at the top of the vertical pipe, or if it is a slide valve it is placed past the quarter-bend which follows the vertical pipe.

Sometimes there is no vertical pipe, but simply a large radius quarter-bend instead. In that case the stop valve comes next after the bend. No matter how placed, the broad principle to be obtained is that the valve shall be dry, there being a fall each way to the boiler and to the engine or steam main. This ensures freedom from water disasters, and avoids the loss and annoyance of drains and traps. In the course of the nearly horizontal pipe between the stop valve and steam main, it is a fairly common practice to fix an automatic non-return valve, which is intended to safeguard a boiler should it be out of work. This valve prevents the boiler from absorbing steam from the other boilers should its pressure fall when cleaning, etc. It also prevents escape of steam should any failure of a boiler tube take place, and confines any escape of steam to the one boiler.

The idea of this valve is excellent, but as usually made with a large rolling ball it is probable that if called on to act suddenly the ball would shatter the valve box, and the last disaster might be worse than the first. These check or non-return valves must be applied with caution.

In the improved form made by Templer & Ranoe and described in the Chapter on "Valves," the moving of a heavy weight through a long distance is avoided.

Danger may arise even when a pipe is being drained of water preparatory to opening the steam valve, especially if the water has become cold. Thus, a horizontal length of pipe below a main steam pipe may become full of cold water, and when partly drained so that the water-level is below the crown of the pipe and steam can enter above the long horizontal surface of the water, there will be sudden condensation of some of the steam; waves will be set up, and inevitably there will be water hammer and probably burst pipes.[1] The fact that a water hammer takes effect chiefly at a valve, a bend, or a tee piece, emphasises the badness of the practice which permits such valve bodies or tees to be made from cast iron. In laying out a pipe system, therefore, the principle to be observed is that of a steady progressive fall from the boiler stop valve to the engine. In addition to this, the question of elasticity must also be fully considered. No absolute fixed plan can be given, because the system must be varied to fit the boilers and their position relative to the engine. In a bank of Lancashire boilers, when the steam outlet is central, the valve is sometimes placed directly upon the mounting block, and the steam branch curves out from one side by a quarter-round bend, and thence proceeds to the rear of the boiler at a height but little above the crown of the boiler, avoiding the manhole because of the lateral

[1] *See Manchester Steam Users' Association.* Memorandum by Chief Engineer, June, 1901.

bend. Without this bend the pipe would be inconveniently close to the manhole cover, and for this case the vertical branch is employed, the valve being high up and upside down (*see* p. 120). The horizontal branch connects to the steam main, which is supported by hangers, or by brackets, on the rear wall. The branch pipes from the boilers may enter on the side of the main, or by a downward bend. The length of these branches, often 15 to 20 feet, combined with the length of the vertical pipe, suffice to afford sufficient elasticity to take up stresses of expansion.

Ranges of Lancashire boilers have frequently been fitted with steam main closely attached to the side outlets of the stop valves. This is an arrangement which provides too little elasticity and is not at all to be considered, especially for high pressures and temperatures.

CHAPTER VII

Pipe Joints

THERE are a wide variety of means of connecting pipes, though these may be classed under three main headings, as follows :—

(1) **Spigot and Socket.**

(2) Screwed and socketed, or flush-jointed.

(3) Flanged.

The first named is named only to condemn the spigot-joint as unsafe for steam or hot water. It will draw apart should its supports fail, and **should not be used.**

The second class is only to be used with sockets, and not in the flush-jointed form. It has been sufficiently referred to under the head of "Steel Pipes."

The third class is found in many forms. Flanges are—

(*a*) Cast with the pipes, whether these are cast iron, cast steel or cast malleable.

(*b*) Welded on to mild steel or wrought-iron pipes.

(*c*) Screwed on to mild steel or wrought iron and sometimes partly brazed in addition.

(*d*) Riveted on to mild steel or wrought iron or copper.

(*e*) Brazed on to copper pipes.

(*f*) Loose upon the pipes, which are gripped together by lips turned up on the ends of the pipes after the flanges are slipped on.

Classes (*a*) to (*e*) are referred to sufficiently under other headings, except as regards dimensions of flanges.

Class (*f*) are too numerous fully to describe. A large number will be found illustrated in a paper read by Mr. R. E. Atkinson.[1] Essentially the loose flange joint is formed by slipping a flange over the pipe, which is afterwards turned outwards to form a lip or small flange, or a thick ring flange is welded on to the pipe ends.

Sometimes each pipe end, if of copper, is flared out to an approximate quarter-sphere, and the loose flanges draw the two ends upon a solid interior joint ring, the outer face of which is an approximate half-circle and is turned with circumferential **V** grooves, into which the copper pipe is forced by the flange pressure.

Copper pipes are perhaps more suitable than steel for loose flanges, especially in small sizes. Thus the 2-inch solid-drawn copper water-control

[1] *Minutes of Proceedings, Inst. M.E.*, 1901, page 443.

pipes of the Cruse Controllable Superheater (Fig. 15) are joined into a continuous spiral by loose thick stamped steel flanges, slipped loose on to each end of the U-pipe. The end is turned over to form a narrow flange, and this flange is drawn up to the face of the connecting link, the copper itself forming the metal to metal joint under the heavy bolt pressure. For various forms of loose flange joints the lists of the Mannesmann Co. may be consulted.

To secure steam tightness between flanges various expedients are resorted to.

The Manchester Steam Users' Association recommend faced flanges with merely a little red paint.

Oiled brown paper is sometimes used on faced joints.

Mr. Dewrance recommends scraped surfaces for pressures of 350 pounds, such as he used.

Ordinary good practice uses woodite rings with flat-faced flanges.

The Babccck Co. use the corrugated copper gasket, the flanges having a projecting face to take the rings.

Compound rings of copper and asbestos have considerable elasticity, and make good joints. The author recommends solid rings of copper wire, and has used ordinary $\frac{1}{16}$-inch copper wire with the ends simply crossed to form a joint ring. What is wanted to secure sound work is a truly faced flange free from spongy metal, and a closely pitched circle of

stout bolts in a strong flange so as to ensure a tight nip on the copper wire. If pipes are pulled apart at any time, the old rings must not be put up again unless they are first heated to a dull red heat and dropped into water to soften them, but old rings become flattened, and it is better not to practise such economies unless the flattened rings can be turned on edge to present their long axes to the flanges.

Solid copper rings are made in the form of two flat V's placed back to back with the idea that the sharp edges of the V-grooves will make a good joint; but as round copper wire is safe and reliable, there seems no good reason to seek further complexity.

Superheated steam obviously demands that nothing of an organic nature shall enter into the composition of a joint ring, for the temperature will soon carbonize it.

Where flanges are weak a joint covering the whole surface must be employed, or the flange may break when the bolts are tightened up. Flanges as shown in Figs. 11, 12, 14, 16 must therefore be strong and stout, to withstand the bolt stress, and when a joint is made with a simple ring of copper wire it should not be attempted to use a light flange. If the flange is light the wire should not be too far inside the bolts, though too large a ring of copper means that more pressure is required to squeeze the ring to a tight joint.

For superheated steam, if not too high in tempera-

rubbed in graphite would make a good joint over a fully-faced flange. The graphite would prevent the ring from sticking to the metal.

Mr. Venning has used Rainbow packing for insertion between faced flanges up to 180 pounds in several of the largest power stations in England.

CHAPTER VIII

Supports

PIPES may be variously supported, as follows :—

(1) By pillars from below, as in Fig. 29.

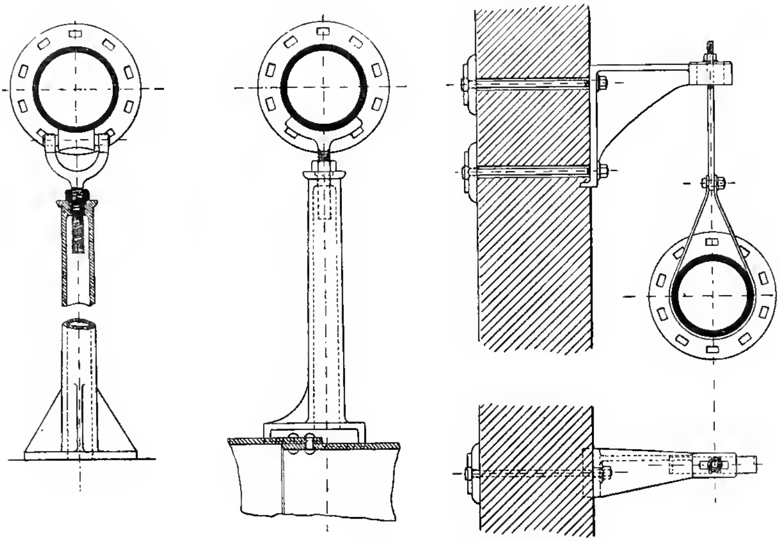


FIG. 29.—PIPE PILLAR AND SUSPENDER (YATES & THOM).

(2) By hangers from above, as in Figs. 29, 30, 31, 32.

(3) By brackets on which the pipe rests (Fig. 33), which shows simply the cross-section of a bracket

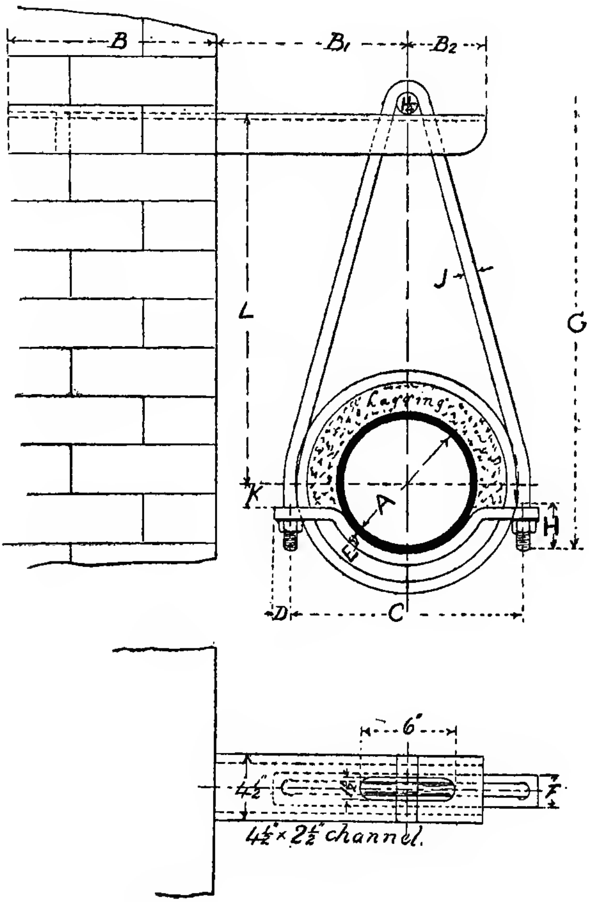


FIG. 30.—PIPE SUSPENDER (BRITISH ELECTRIC TRACTION CO.).

of the form of Fig. 31 carrying the pipe on its upper table.

(4) On brick piers.

In the pillar form (Fig. 29), which is convenient for carrying pipes from the crown of a boiler or from the brick walls of the seating, the slightness of the pillar affords play for expansion movements. The head of the pillar is arranged with an adjustable

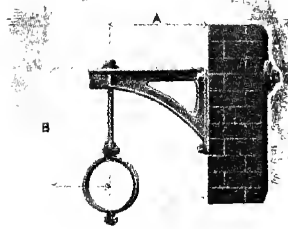


FIG. 31.—SUSPENDER FOR ONE PIPE (BABCOCK & WILCOX CO.).

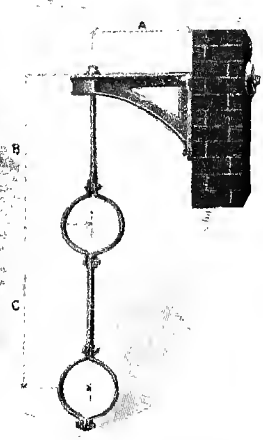


FIG. 32.—SUSPENDER FOR TWO PIPES (BABCOCK & WILCOX CO.).

screw to enable the weight of the pipe to be carried without either undue upward or downward stress on the connections of branching pipes. The part on which the pipe rests is a bow of about 90° arc, to which the pipe must not be tied down unless the bow is merely fitted into the pillar by a tail free to move up in case of the pipe rising frcm expansion.

Fig. 29 shows also the suspender of Messrs. Yates

& Thom, which may be slung as shown from a wall bracket or from an overhead girder. The upper nut may have a helical spring placed between it and the bracket. When adjusting any form of pipe carrier with branches, the latter may be unbolted from the main pipe and their weight carried by a rope and balance weight equal to half the weight of the supported pipe if this is fastened to the rope near one end. The main pipe is then adjusted to height and the branches bolted to it. A little upward stress may be given when cold, as this will be relieved when hot by the expansion of the vertical branch on the boiler, if present. Suspended pipes are as free to move as their various attachments will permit. Often they will be set swinging or vibrating by the pulsating action of the draught of steam by the engines. This is usually best attended to after setting to work. It may usually be checked by lightly wedging the pipe at a point where one of the branch pipes passes through the wall. If the wedge only stops the full amplitude of the vibration, this may usually be entirely stopped.

When pipes are simply supported on brackets they will not vibrate so readily.

Brackets are often hollowed to the curve of the pipe, but this is a doubtful advantage, tending to prevent lateral movement under the push of the branch pipes.

It is better to provide the table of the bracket quite flat and plain. Riveted to the pipe or fastened

by a pair of clip-rings encircling the pipe, there should be a rubbing piece of iron interposed between the pipe body and the bracket to take up the wear due to constant movement. These rubbing pieces should be from $\frac{3}{8}$ to $\frac{5}{8}$-inch thick. If a pipe is to be anchored at any bracket the rubbing piece may be as per Fig. 33, with down projecting pieces straddling the bracket table loosely, and with lateral extensions *a a* to take the clips *c c* firmly holding the rubbing piece to the pipe. The encircling clips are in halves, the bolted ears being placed at a convenient angle, preferably horizontally.

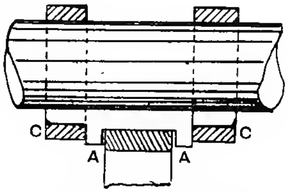


FIG. 33.—PIPE BRACKET.

Pipe supports are intended only to carry weights and should not be arranged to prevent free movement necessary to relieve the expansion stresses, except of course at such point as is intentionally selected as an anchorage. In the type of hanger of Fig. 30, with girder and wrought-iron rods and clips, the girder is simply a piece of $4\frac{1}{2}'' \times 2\frac{1}{2}''$ channel built into the wall, and the pipe is carried by a saddle slung by a double-ended nutted suspender, slung over a pin carried on the girder. This is the design of the British Electric Traction Co., and the dimensions are given in the accompanying table.

TABLE XVII.

Table of Dimensions of Pipe Suspender (Fig. 30).
For Steam and Lagged Pipes.

Dia. of Pipe.	External Dia. of Pipe A	Length of Channel.			Size of Strap.				Dia. and Length of Sling.				
		B	B1	B2	C	D	E	F	G	H	J	K	L
in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.
2	2½	10	8	4	8½	¾	½	1½	16	2½	½	⅜	13
3	3½	10	8	4	9½	¾	½	1½	17½	2½	½	½	14
4	4½	12	9	4	10½	1	⅝	1¾	18	2¾	⅝	¾	14
5	5⅝	12	10	5	11½	1	⅝	1¾	20	2¾	⅝	1	16
6	6⅝	12	10	5	12½	1	⅝	2	22½	3	⅝	1¼	18
7	7⅝	12	12	5	13½	1¼	⅝	2	24	3	¾	1½	19
8	8¾	14	12	5	14	1¼	¾	2	25	3	¾	1½	20
9	9¾	14	13	5	15½	1¼	¾	2	25	3	¾	1½	20

For Exhaust and Bare Pipes.

Dia. of Pipe.	External Dia. of Pipe A	Length of Channel.			Size of Strap.				Dia. and Length of Sling.				
		B	B1	B2	C	D	E	F	G	H	J	K	L
in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.
3	3⅞	10	8	4	5½	¾	½	1½	15½	2½	½	½	12
4	5	12	9	4	6¾	1	⅝	1¾	15¾	2½	⅝	¾	12
5	6	12	10	5	8	1	⅝	1¾	18	3	⅝	1	14
6	7	12	10	5	9	1	⅝	2	20½	3	⅝	1¼	16
7	8⅛	12	12	5	10½	1¼	¾	2	21	3	¾	1½	16
8	9⅛	14	12	5	11½	1¼	¾	2	22	3	¾	1¾	17
9	10⅛	14	12	5	12½	1¼	¾	2	22½	3	¾	2	17
10	11¼	14	15	6	14	1½	¾	2	24	3	1	2¼	18
11	12¼	18	15	6	14¾	1½	1	2½	24½	3	1	2½	18
12	13½	18	17	6	16	1½	1	2½	27	3	1	2¾	20
13	14½	18	18	6	17½	1½	1	2½	30	3½	1	3	22
14	15½	18	19	6	19	1½	1	2½	30	3½	1	3¼	22

The resting of pipes on rollers carried by brackets is not usually thought so good as the plain rubbing contact which introduces an element of stability against vibration; but after all, the roller-carried pipe is less liable to vibrate than is the slung pipe. Rollers are, however, liable to become set fast, and they are then liable to wear the pipes which have not perhaps been supplied with rubbing pieces.

In the type of bracket of Fig. 29 there should be a projecting lug at the bottom of the wall plate to rest in the wall for the purpose of taking the weight. The two top bolts must be strong enough to carry the load acting with an intensity of pull on the bolts $S = \frac{W \times A}{N}$, where N is the distance between the top bolt and bottom of the bracket and A is the distance from the wall to the pipe centre, W being the weight of pipe. The heads of the bolts must be carried by back plates, which may be either simple double-hole washers, or a full plate, as large as the wall back of the bracket.

The Babcock Co. make brackets, as in Figs. 31, 32, the dimensions and weights of which are given in the annexed Table XVIII. Brackets of plain bent angle iron with a riveted jib piece of flat iron, are made with the dimension A reduced to about half the length, and B and C to less than half for small pipes, and to three-fifths for larger pipes, and they weigh less than a third of the cast-iron brackets.

TABLE XVIII.

PIPE HANGER (BABCOCK & WILCOX CO.).

Suitable for Pipes of Internal Diameter.	3 in.	4 in.	5 in.	6 in.	7 in.	8 in.	9 in.	10 in.	12 in.
Dimension A . . .	2′ 0″	2′ 0″	2′ 0″	2′ 3″	2′ 3″	2′ 3″	2′ 6″	2′ 6″	2′ 6″
Dimension B . . .	2′ 2¾″	2′ 3¾″	2′ 4⅛″	2′ 4⅝″	2′ 5¼″	2′ 6″	2′ 6½″	2′ 7¼″	2′ 8¼″
Weight, (approx.) lb. . .	121	126	126	151	151	162	177	177	177
Weight, (approx.) kilos. . .	55	57	57	68	68	73	80	80	80
Dimension C . . .	2′ 5½″	2′ 7½″	2′ 8¼″	2′ 9¼″	2′ 10½″	3′ 0″	3′ 1″	3′ 2½″	3′ 4½″
Weight, (approx.) lb. . .	12	135	135	160	160	177	192	192	192
Weight, (approx.) kilos. . .	57	61	61	72	72	80	87	87	87

Pipes may be carried by simple projecting girders, similar to that shown in Fig. 30, the pipe being fitted with rubbing pieces, as in Fig. 33, and where required with anchor-plate also, as in Fig. 33.

CHAPTER IX

Erection of Pipes

IT is very usual in putting up a system of pipes to leave certain gaps to be filled on completion with making-up lengths. These usually cause delay in completing a piece of work.

Carefully set-out work, erected to exact dimensions, as it may be, may have all piping ordered from the start, so as to fit without making-up lengths.

In either case, if the pipes are to have an initial stress when cold equal and opposite to their stress when hot, due allowance must be made to meet this requirement, and the final bolting-up of the last piece will be easier effected after steam has been got up and the pipes blown through and heated, which may be done by blanking the ends still open and letting in full pressure, with due regard to unbalanced stresses. When the last pipe is difficult to complete and it is obvious that when hot the cold stresses will be relieved more or less, it is desirable to loosen out several flanges so as to distribute the final gap among several flanges, which are to be all screwed up together a little at a time.

It will sometimes happen that the final pipe does

not present its flange parallel with its fellow. This fault will probably not be remedied when at working temperature. It may be treated drastically by building round it in place a coke or charcoal fire in a fire-basket, so as to heat the pipe over a length of three or four feet. While hot the flanges may be tightened up and the pipe will take a set, and should be left to cool slowly, filling the fire with lime or wood ashes to ensure this.

This fault of non-parallelism of flanges is often due to faulty methods of templet making.

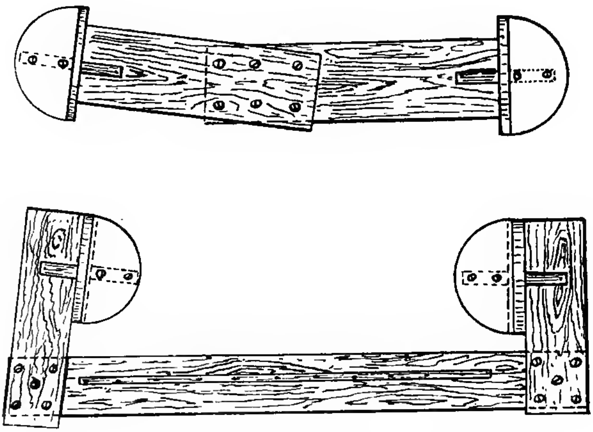
A pipe templet consists of a flat board, to which are attached wooden flanges. The wooden flanges are fixed to the ends of the pipes to be joined up, and the board between is then screwed firmly solid, care being taken that no stress is set up. When released the flanges should fit the flanges they are to meet when finally cast. Usually this templet is sent to the foundry. This is a mistake. It should be kept, and a second templet made from it, but reversed, should be sent to the founder.

Thus Fig. 34 shows the first templet. To this is fitted the second, Fig. 35. The flanges of this latter are then convenient against which to fix the flange patterns in the sand. This cannot be done with the first templet of Fig. 34, and the use of the second templet ensures more accurate make-up lengths than can be got by the usual method.

Similar templets must be made to send to the steel pipe makers.

For cast pipes the first templet must be made up

in length by an amount equal to the contraction of the casting, plus the amount lost in facing, less the joint rings and the gap for initial stress. Usually the templet as made will produce a pipe quite as long as will more than fill the length when at working temperature. These matters must be decided before making the templet, and if the final pipe is to be shorter than the above will produce, the gap



FIGS. 34, 35.—TEMPLETS FOR PIPES.

to be filled can first be shortened by a parallel blank wooden flange.

Every endeavour should be made to avoid the final necessity of preparing a taper joint ring to fill the taper gap of non-parallel flanges, and where possible works must be set out correctly and erected to plan, so that the pipes can be ordered from the first.

This demands correct work on the part of the

engine builder and the fixing of a datum or reference line as between the engine and boiler department, so that the two contractors cannot shirk the responsibility. By some it is thought to be impossible to measure up for pipes correctly or to set up engines and boilers so that the pipes made to drawing will come into place, but with care this can be done.

Adding to Existing Pipe Systems.

It is sometimes desirable to add a branch line to an existing pipe system with a minimum of stoppage. This can be done without taking out any existing pipe for the purpose of putting in a junction tee. Instead of this a saddle is prepared, as shown in Fig. 36, which fits upon the pipe. This saddle is jointed by insertion, or other suitable material, according to the pressure, and is gripped upon the pipe either by a similar flanged saddle and bolts or by strong U-bolts, with nuts on the saddle flanges. After the saddle is firmly fixed and provided with a shut-off valve that can be quickly bolted up, the pipe may be emptied of steam, and an opening cut in it by a drill, or a leading drill followed by a crown, cutter. This done the valve is bolted on, steam admitted, and any residual cuttings blown out through the valve. Where it is not possible to shut steam off for even the short period indicated, the valve must be of the fullway type, blank flanged with a special stuffing gland-head, through which

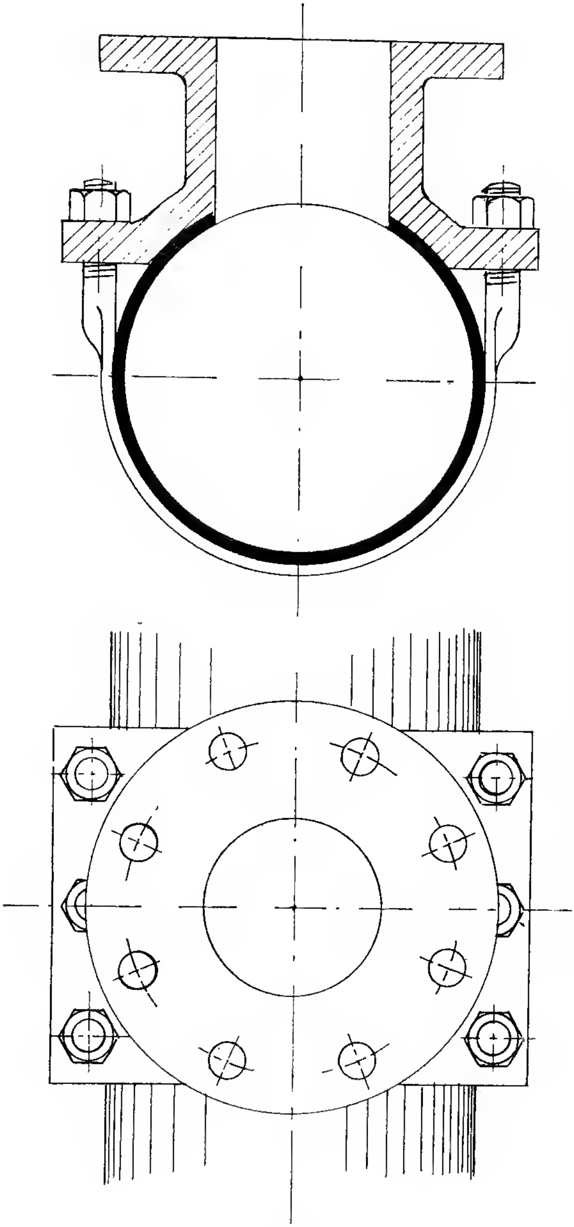


FIG. 36.—SADDLE FOR ADDING BRANCH PIPES TO STEAM MAIN (VENNING).

works the spindle of the drill and cutter. In this case a small valve ought to be fitted to the body of the shut-off valve, through which the cuttings of the drill can be discharged as the work proceeds. The continuous flow of steam will prevent any cuttings from falling into the main steam pipe and will keep the cut-out disc upon the leading drill, so that on completion of the cut the disc can be drawn back through the valve, the valve closed and the drilling fixture removed. The author is indebted to Mr. A. Venning for this saddle connection. When the conditions permit of it the hole cut through the pipe may be merely a number of drilled holes arranged so as to leave a portion of solid plate, though it will be observed that the saddle itself acts as a reinforcement of the weakened pipe, and that any weakening is upon so short a length as to be of little account. Still it should be considered, and if necessary provided against.

Before proceeding to erect pipes, all the bolts should be cleaned by boiling in soda, oiled, and the nuts run over until they can be turned with the fingers. As each pipe is put in place it should only be loosely bolted, using a short spanner easily. Steam should be sent through to heat up the pipes, and all bolts gradually and in turn tightened up, first with medium spanner, or spanners held at half-length, and finally with full-length spanner. The steam pressure should be only a pound or two above atmosphere during this work. If any joints leak when steam is got up to full pressure, they should

be marked and attended to after pressure is shut off. The risk of leakage will be minimized by the gradual method outlined above.

Pipes of small size may be bent cold, but a better job is usually made by heating to a clear red and bending round pins fixed on a bench or in the vice. A good workman will often bend an empty pipe without spoiling its circularity by coaxing it in the vice, with which he squeezes it laterally and prevents it taking an oval section round the bend. Less experienced men should fill the pipe with sand, thoroughly dried, to prevent explosion by steam. Such a filled pipe will remain truly circular when bent, and the sand will drop out as soon as the end plugs, which retain it, are removed. Copper pipes may be filled with melted resin, which when cooled will enable the pipe to bend without crippling, the resin being plastic-brittle, and crushing round the bend with expansive effect. The resin must be melted out. Or pipes are filled with melted solder or other alloy of low melting point. Water or oil would probably serve if the ends could be kept tight. The theory of bending by using a filling is of course that a circle contains the maximum area of cross-section for a given periphery, and any change from a true circle would have to compress the filling material. It is easier for the pipe to maintain its full circle. An unfilled pipe is apt to cripple on the inside of the bend, or to flatten on the outside. Iron or steel pipes can always be made sufficiently hot by laying them in a long brick trough built up of

loose bricks, with air spaces and with space under the pipe for fire. Wood or coal can be used as fuel. The acquisition of the clear red temperature can best be judged in an unfilled pipe by looking through it as it lies in the fire. Above four inches pipes can rarely be bent or shaped except by pipe-makers.

CHAPTER X

General Arrangements

THE general arrangement of a modern power station demands that the steam pipes enter into consideration as a prime factor, and not as an after-thought. This is especially the case where superheat is to be employed.

The common arrangement of a long row of boilers separated from a long row of engines by a wall, with sometimes a steam main on each side of the wall, is by no means good. Ring mains are bad practice, yet in such a design it becomes almost imperative to adopt a form of ring main if separately fired superheaters are to be employed, for steam must be led to the superheater from all the boilers in a section, and from the superheater to the engine main. In the face-to-face arrangement of boilers such as was adopted for the Central London Railway or the L.C.C. Station at Greenwich, the placing of a separately fired superheater almost compels a ring main, and such stations are examples of an unfortunate shape of the plot of land on which they are built. With steam turbines capable of using more

steam in a given length of engine room, the disposition of boilers must be along a line at right angles to the engine room, i.e. each boiler is parallel with the length of the engine room, and the bank of boilers is transverse thereto.

The number of boilers in each bank will depend upon the demands of the engines, and usually would be such as are necessary to supply the demands of the engine. There is a tendency to regard each engine and its boilers as a separate entity.

This would preclude all idea of steam mains other than that of coupling all the boilers in a bank and extending to the engine. Where it is considered necessary to have one engine at work and another moving slowly round in case of mishap, this idea would also imply a set of boilers at work and a second set under full pressure with banked fires, and in a considerable installation this would be uneconomical, and two adjacent engines may at least be able to draw steam from one set of boilers. It is certainly not good economy to multiply boilers to the extent that the full carrying out of this idea would demand. The total boiler capacity need only be more than the mean demand for steam in a tramway station by the amount of the necessary spares and the number of boilers in each bank, and the number of banks will, therefore, depend upon the total number required, the length available and the superheater dimensions.

The requirements of the superheater demand to be considered, and if the banks of boilers are placed

at right angles to the engine room this enables the separately-fired superheater to be placed next the engine-room wall between it and the first boiler. All the boilers turning their steam into a main pipe run across the boilers, this pipe is carried directly forward to the engine room with a stop valve in the longitudinal centre line of the superheater. On each side of this stop valve branch out the pipes which form a loop with the superheater. Each end of the loop has its valve close to the steam main, and the engine may thus be supplied with steam saturated or superheated by suitable opening or closing of these three valves.

This arrangement minimizes the length of pipe containing superheated steam : it affords the simplest and most direct run for the steam from each boiler to the engines. If a steam main is wanted in the engine room, it can easily be arranged to take steam from each superheater and deliver it by branches to the engines, but this main should be proportioned rather as a balancer than as a main to carry all the steam made at any point beyond a given section to any other point in the opposite direction. If too much is thought of providing every possible permutation and combination of steam boilers and engines, the inevitable result will be that an excessive diameter and cost of pipe will be entailed. It is sounder practice to use small pipes where improbable combinations are to be provided for and allow temporary loss of pressure by wire-drawing, than to charge standing

expenses with the extra interest on heavy pipes and the excessive cost of heat radiation losses.

The steam pipe across each bank of boilers may for economy be stepped down in diameter towards the extreme end. Where this is done, and there is a prospect of extensions being made, it is obvious that this ought theoretically to take effect between the superheater and the first boiler. The same result will be obtained by putting in the new boiler at the extreme outer end of the bank, and putting in a new and perhaps larger length of steam pipe at the opposite end. This points to the necessity of accurately spacing all boilers alike and having each section of boiler main pipe of exact length, so that the boiler branches will fit the main when this is moved endwise a section or two more. Usually this will not be the mode of extension, though it might follow the introduction of an improved form of turbine. Should occasion demand it, each boiler main pipe may be connected across to the pipe of the next bank in order that an excess of superheaters need not be installed, but usually a well-managed station can be worked so as to enable the cleaning of a superheater to coincide with the cleaning of several boilers in one bank, and the virtual stoppage of the whole bank.

This should be aimed at by providing rather a small amount of first-class plant than an excess of cheap trash.

As an illustration of the practice of economy in pipe dimensions, suppose for simplicity that half

the plant in a station was spare. Then the worst condition for the steam main along a row of engines would be that the whole of the north engines were supplied from the south end boilers, all the south engines and north boilers being at rest.

Suppose this demanded the middle of the steam main to have an area of 400 in. cross-section to give the proper velocity of flow of steam. A good practical solution with a view to probability and economy would be first to cut the area to one-half on the assumption that not more than one-half the total steam would be called on to pass any one point in the main, and secondly to reduce it still further by perhaps 20 per cent. on the assumption that under conditions of such rare occurrence the velocity of the steam might be increased rather than that the station should carry a perpetual interest charge on huge pipes and a heavier radiation loss.

Designers may also consider the use of reduced sizes of valves where economy in cost is of more than usually serious moment. The resistance of pipes is so much a matter of length that a short piece of smaller size may be permitted, and smaller valves of the fullway type with conical ajutage will serve to cut down expenses.

Thousands of pounds have been wasted on excessive valve proportions in excessive mains which are never called upon to give other than a balancing effect. One of the evils of the ring main is that it must be equally large throughout, for it is only put up upon the assumption that it must carry all the

steam any way round. It is crowded with valves, supposed any one of them capable of being moved if a mishap occurs. Yet so little is the design of them thought out that the handles of the valves may be found carried up to platforms placed directly over the mains, so that should an accident happen the man who essays to close the valves will be scalded by the steam. Steam is lighter than air in the ratio of 9 to 15 for equal temperature, and valves should rather be got at below the level of possible steam escape.

Similarly in a bank of eight boilers, of which there will usually be six at work, the size of the pipe must nowhere be larger than required to carry steam from six boilers, and this size will extend from No. 6 past Nos. 7 and 8. Indeed, it is open to argument that the 5-boiler size might be extended to between Nos. 6 and 7, for it would usually be contrived not to have 7 and 8 off at the same time, and only steam from five boilers would usually traverse the section of pipe up to No. 7. In the odd event of doing so, it would be permissible to allow temporary increase of velocity of the steam. Such, then, are the principles affecting design where proper consideration is given to economy of coal and maintenance and running expenses, so as to avoid rendering the cost of connecting up the main items of plant greater than the cost of the plant itself.

Power-station designers should also be prepared to consider the question of getting very much more steam from boilers than has hitherto been attempted.

The author has long advocated the use of feed water heated fully up to the boiler temperature, and has adopted the conservative estimate of a better output by 20 per cent. as the effect to be expected from fully heating up feed water in the control system of a superheater. Mr. Cruse supports this estimate, but Col. Crompton has stated that with such a fully heated feed a water-tube boiler has evaporated at least double its ordinary rated output. There is at least evidence that boilers, badly worked as they usually are, are in excess of what they need be in better practice.

This question, however important to the design of pipes, is, however, too large to be further dealt with here. The point has an important bearing, however, on the whole question of pipes, both in general arrangement and in the dimension of the pipes of each individual boiler.

CHAPTER XI

Valves

SOME valves are always necessary in the course of a steam pipe, but the number should be a minimum. One valve at least must always be applied, viz. the boiler stop-valve, the position of which has been discussed elsewhere.

As with junction pieces, valves are exposed to the stresses in a system of pipes, and are often made with cast-iron bodies even in a system of steel pipes. Thus in Figs. 37 and 38 are shown two types of valves by Templer and Ranoe, which are supplied with cast-iron bodies for pressures up to 200 lb. The valves and seatings are of high pressure bronze, the seating being fixed into the body by studs, as shown, thus avoiding the warping of the seat by differential expansion, such as occurs where seats are tightly pressed into cast iron. The joint is made on the flange, and admits of a sufficiency of movement to take up expansion. Tables XIX., XX. give the leading dimensions.

In these valves the spindle has a head which makes a free turning joint on the valve. As in all good practice the screw of the spindle is outside

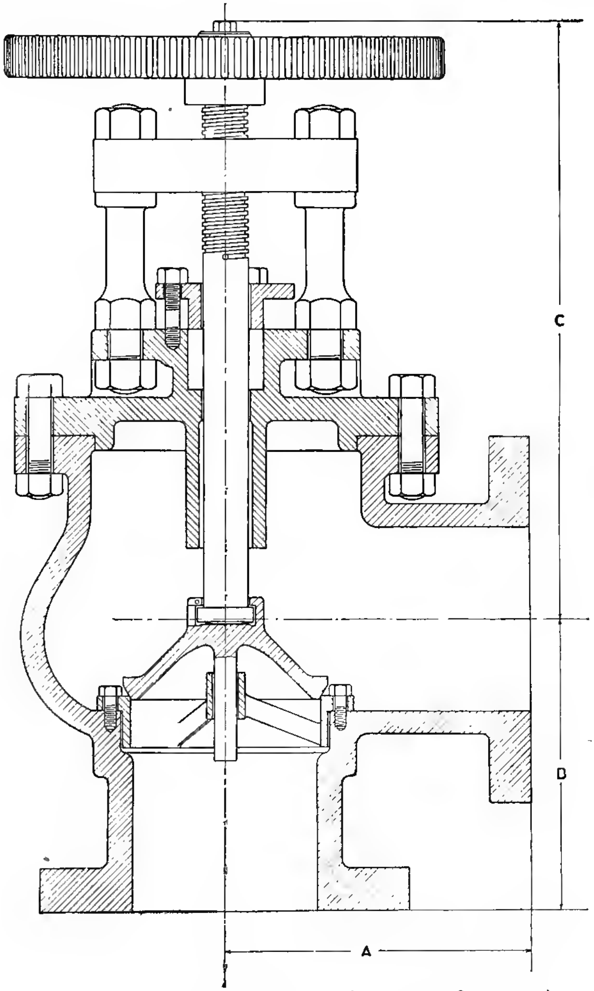


FIG. 37.—ANGLE STOP VALVE (TEMPLER & RANOE).

and carried in a cross bar with pillars on the cover. The screw is thus fully in sight, and can be kept

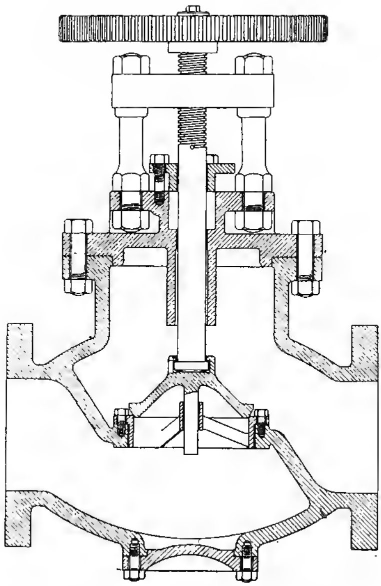


FIG. 38.—STRAIGHTWAY STOP VALVE.

TABLE XIX.

Cast-Iron Bodies and High-pressure Bronze Working parts for Steam Pressures up to 200 lb. per square inch.

RIGHT-ANGLED PATTERN (FIG. 37).

Bore . .	2	$2\frac{1}{2}$	3	$3\frac{1}{2}$	4	5	6	7	8	9-in.
Dia. of Flanges	$6\frac{1}{2}$	7	8	$8\frac{1}{2}$	9	$10\frac{1}{2}$	12	$13\frac{1}{2}$	15	16 ,,
Thickness of Flange	1	1	$1\frac{1}{8}$	$1\frac{1}{8}$	$1\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{3}{8}$	$1\frac{1}{2}$	$1\frac{5}{8}$	$1\frac{3}{4}$,,
Dimension, B . .	$4\frac{1}{2}$	5	6	7	$7\frac{3}{4}$	$8\frac{3}{4}$	$9\frac{1}{2}$	$10\frac{1}{2}$	$11\frac{1}{2}$	12 ,,
Dimension, A . .	$5\frac{3}{4}$	$6\frac{1}{2}$	7 $\frac{5}{16}$	$7\frac{1}{2}$	8	$9\frac{1}{4}$	10	11	12	13 ,,
Dimension, C . .	12	$12\frac{1}{2}$	$12\frac{3}{4}$	$13\frac{1}{4}$	17	$18\frac{5}{8}$	$19\frac{3}{4}$	$21\frac{1}{4}$	$22\frac{3}{4}$	$24\frac{1}{2}$,,

All high-pressure gunmetal for 200 lb. pressure.

Bore	2	2½	3-in.
Dia. of Flanges . .	6½	7	8 ,,
Thickness ,, ,, . .	⅝	¾	13/16 ,,
Dimension B . . .	4½	5	6 ,,
,, A . .	5	5⅜	5¾ ,,
,, C . .	8¼	10¼	11 ,,

TABLE XX.

Straightway Pattern (Fig. 38).

Cast-Iron Bodies and High-pressure Bronze Working Parts for Steam Pressures up to 200 lb. per square inch.

Bore . .	2	2½	3	3½	5	5	6	7	8	9	10	12 in.
Dia. of Flanges.	6½	7	8	8½	9	10½	12	13½	15	16	17	19 ,,
Thickness of Flanges	1	1	1⅛	1⅛	1⅛	1¼	1⅜	1½	1⅝	1¾	1¾	2 ,,
Length over Flanges	10	11	12	13	14	17	20	22	24	26	28	32 ,,

All high-pressure gunmetal for 200 lb. pressure.

Bore . . .	2	2½	3-in.
Dia. of Flanges. .	6½	7	8 ,,
Thickness . .	⅝	¾	13/16 ,,
Length over Faces .	8	9	1 ,,

clean and lubricated. Fig. 37 is suitable for a boiler stop-valve when placed directly on the mounting block or on the top of a vertical branch from that block. The outlet flange may point either longitudinally along the boiler or transversely as circumstances demand. In the latter

event there is introduced a quarter-bend between the valve and the straight branch to the main, and this may be desirable on the score of elasticity.

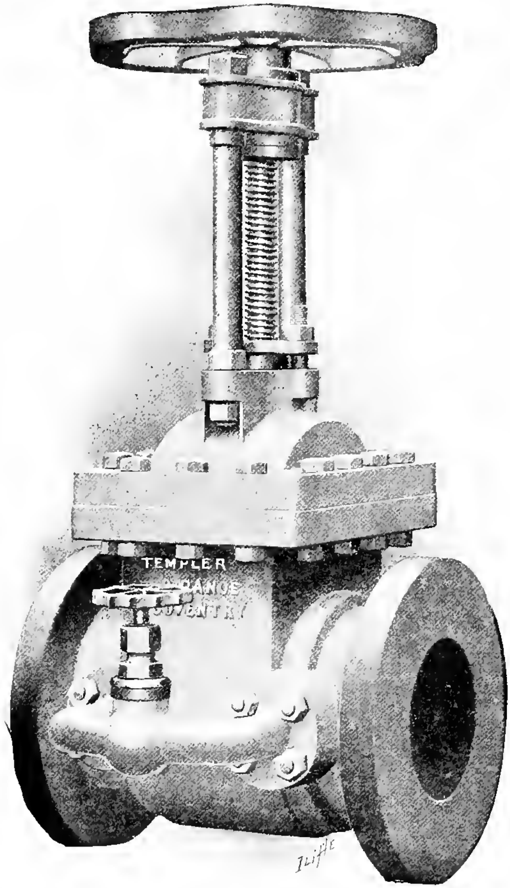


FIG. 39.—FULLWAY MAIN VALVE WITH BYE-PASS RELIEF VALVE.

The figures show how the pillars should be attached —not being simply screwed into tapped holes, drilled often by error or carelessness through into the steam space. When used as boiler stop-valves

there is no difficulty in opening these valves, but where valves are placed under conditions which demand their opening against the steam pressure, it is usual in the larger sizes to provide a smaller bye-pass valve fixed to the body of the valve as

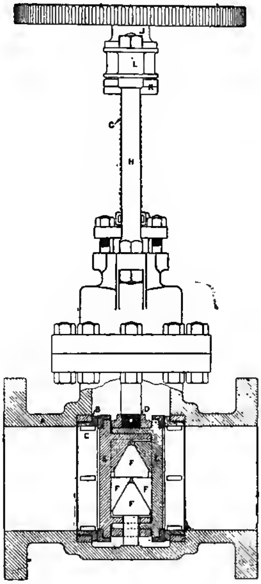
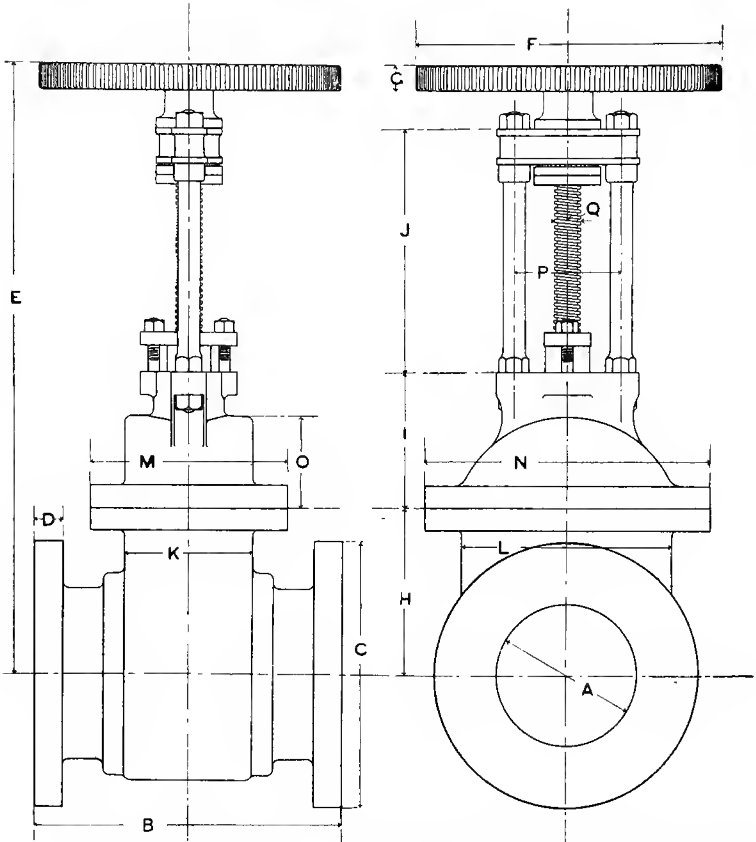


FIG. 40.—SECTION OF "FULLWAY" VALVE.

in Fig. 39, in order that the pressure may be equalized on the two sides of the main valve before opening it. The bye-pass serves for warming-through purposes also. Thus Fig. 39 shows a straightway stop-valve with bye-pass, and in Fig. 40 the valve is shown in section, and diagrammatically in Fig. 41—*A* being the body; *B* the seat ring of the same material forced into the body; *C* the bronze or nickel alloy seat screwed into *B* and making a joint on the flange; *D* the carrier, which lifts or



A	B	C	D	E	F	C	H	I	J	K	L	M	N	O	P	Q
2"	10"	6½"	1	13¼	8"	½"	3¼"	2½"	5⅝	3½	4½"	5⅜"	6¼"	1⅛"	4"	½"
2½"	10½	7	1	13¾	8½"	⅝"	3¾"	2¾	5⅜	3¾"	4¾"	5⅞"	7¼"	1¼"	4	⅝"
3"	11½"	8	1⅛	16¼"	10½"	¾"	4¼	3⅛	6⅜"	3¾"	5⅛"	6¼"	8¼"	2⅛	4¼	1"
3½"	12½	8½	1⅛"	18¼"	11"	⅞"	4½	3¼"	7⅛	4¼"	6⅜"	7⅛	9⅛	2¼	4¼	1
4"	13"	9	1⅛"	19⅞	12"	1"	5⅜"	3⅜	7⅜	4⅛"	7¼"	7⅝	10½	2⅜	5"	1⅛"
4½"	14"	9½	1¼	21⅝"	12½	1"	6⅛	4	8⅝"	4½"	7⅝"	7⅞	10⅛"	2⅝	5"	1⅛"
5"	15	10½	1¼	23½	13"	1⅛	6½	4½	9½	5⅛"	8⅛"	9⅛"	11⅛"	3	5	1¼"
6"	15½	12	1⅜"	26½	14½	1¼"	8	5"	11⅛"	6	8⅝	10⅛"	12⅝"	3⅛	5½	1⅜
7"	17"	13½	1⅜"	30⅝	15½"	1½	8⅝	6⅛	12½	6⅝"	10½"	10⅝"	15½"	4⅛	6	[illegible]
8"	17½"	15"	1⅝	34⅛	17	1⅝	9⅝	7⅝"	14⅝	7¼"	12⅝"	11⅝	16⅛"	5⅛	6	1⅛
9"	18"	16	1⅝"	36⅛	16½	1⅝	11½	7½"	14½	8⅛"	13½"	11⅛	16⅛	4½	6⅝	1⅛
10"	18	17"	1¾	39⅝	20"	1⅝	11⅛	8½"	15⅝	7⅝"	14⅝"	11⅛	18⅛	6	6⅝"	1⅝
12"	20	19	2	45½	23"	1⅝"	14	10⅝	17⅝"	9½"	16⅝	13⅛"	20⅝	7½	6⅛	1⅝
13"	20	21	2	49⅝	24⅛	1⅝	17⅝	8⅝"	18"	9⅝	18⅛"	14½	23½	5⅝"	8"	1⅝
14"	20"	22	2	55⅛	26	1⅝	18⅝	9½	22"	9⅝"	20½"	14⅝	23⅛	6½	9	2

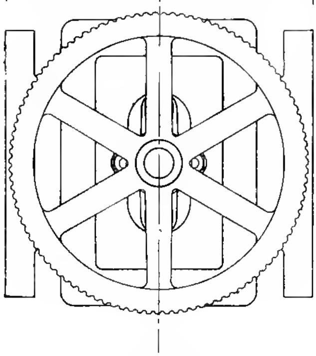


FIG. 41.—" FULLWAY " VALVE WITH GENERAL DIMENSIONS (TABLE XXI.).

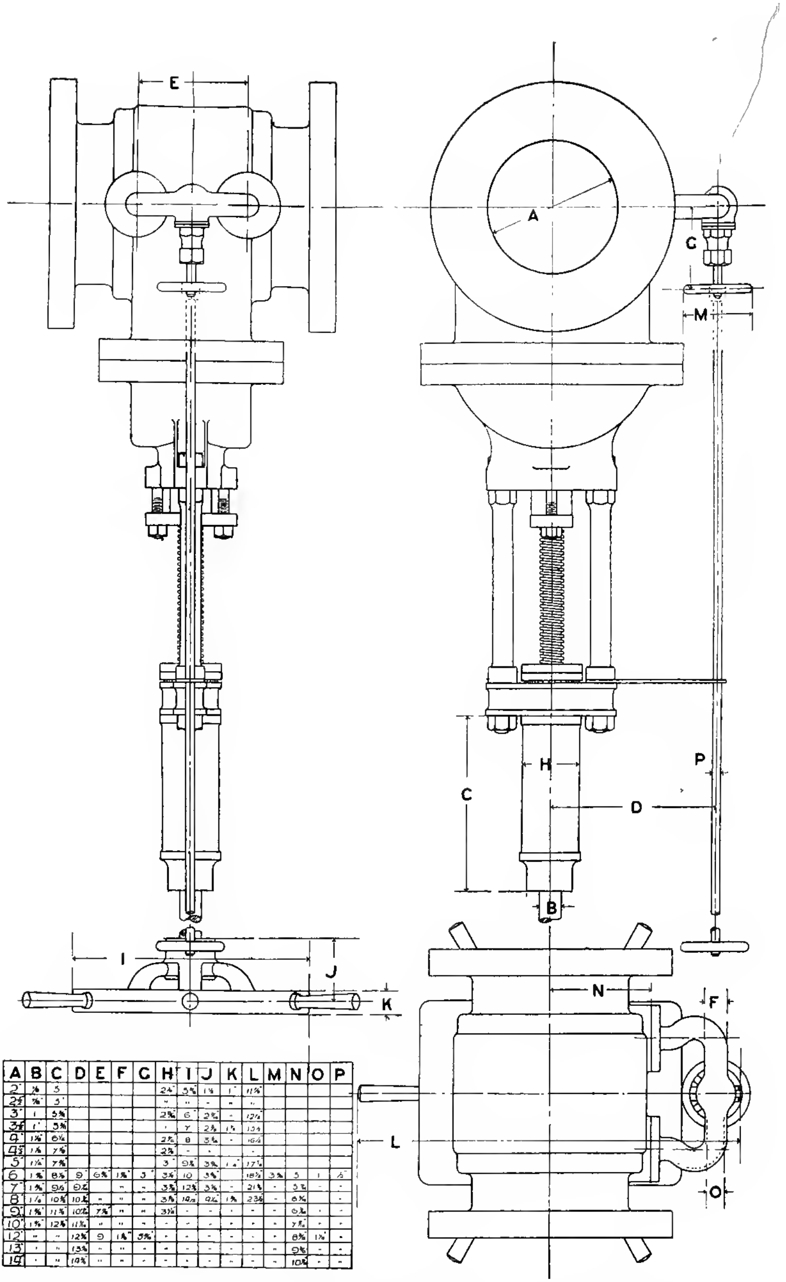
lowers the valve; *E E* the two halves of the valves carrying faces of bronze or nickel alloy for superheat, and *F F* being the wedges which force the valve against their seat when the spindle of the lower wedge *F* touches the bottom of the body casting.

A table of general dimensions of fullway valves (Fig. 41,) by Templer and Ranoe is given on page 121, and will be found fairly approximate for other makes of valves.

The hand wheel *F* may be made into a gear wheel, and turned by a pinion with a long spindle extending downwards within reach, in cases where valves are placed high and out of reach. Such wheels and pinions ought to be made with wide teeth for strength as they do not perhaps afford the same power over the valve as the direct method of Fig. 42 and Table XXII.; which, however, demands a reversed position of the valve with the risk of possible objectionable leakage of water at the gland. The hand wheel of a reversed valve should be attached by a nutted screw, as the wheel may drop off when only keyed.

Valve Position.

Valves may sometimes be seen upon a ring main with their spindles brought up to a gallery, with a grid floor placed directly over the ring main. Obviously, any accident to the main may envelope the valve gallery in steam and render the hand-wheel inaccessible. This arrangement is on a par with the intelligence which employs ring mains by choice.



A	B	C	D	E	F	G	H	I	J	K	L	M	N	O	P
2"	⅞	5					2¼"	5⅝	1¼	1"	11⅝"				
2½"	⅞"	5"					"	"	-	"	"				
3"	1	5⅜"					2⅜	6"	2¾"	-	12¼"				
3½"	1"	5⅞					·	7	2⅞	1⅛	13¼				
4"	1⅛"	6¼					2⅝	8	3⅛	-	16¼"				
4½"	1⅛	7⅛"					2⅞"	-	·		-				
5"	1¼"	7⅜"					3"	9⅜"	3⅜	1¼"	17¼				
6	1⅜"	8⅜"	9"	6⅜"	1⅜"	3"	3⅛"	10	3⅝"		18½	3⅜	5	1	½"
7"	1⅜"	9½"	9¾		"	-	3⅜	12⅜"	3⅝	-	21⅜	-	5¾	-	-
8"	1½"	10⅜	10¼	"	"	"	3⅝"	14⅛"	4⅛"	1⅜	23⅜	-	6⅜		-
9"	1⅝"	11⅜"	10⅞	7⅜"	"	"	3½"	-	-	·	-	-	6⅞	·	-
10"	1⅝"	12⅜"	11⅛	"	"	-	-	-	·	·	·	-	7⅛"	·	·
12"	"	"	12⅜"	9	1⅜"	5⅜"	·	·	-	-	-	·	8⅜"	1⅛"	·
13"	·	"	13⅜	"	"	-	-	-	-	-	-	"	9⅜	-	·
14"	·	"	14⅜	"	"	·	-	·	-	"	-	-	10⅜	·	"

FIG. 42.—" FULLWAY " VALVE REVERSED, WITH EXTENSION SPINDLE (TABLE XXII.).

TABLE XXI.

Dimensions of Fullway Valves (Fig. 41).

A	B	C	D	E	F	G	H	I	J	K	L	M	N	O	P	Q
ins.	ins.	ins.	ins.	ins.	ins.	ins.	ins.	ins.	ins.	ins.	ins.	ins.	ins.	ins.	ins.	ins.
2	10	6½	1	15	8	1	3½	2⅜	6⅞	3⅜	4⅞	6	7½	1 7/16	4	⅞
2½	10¾	7	1	16	8	1	3¾	2⅝	7⅜	3⅜	5¼	6	8	1⅝	4	⅞
3	11½	8	1⅛	18⅞	10	1⅛	4⅜	3⅝	8½	3⅞	6⅛	6¾	9⅛	2 11/16	4½	1
3½	12½	8½	1⅛	20⅝	11	1⅛	4⅞	3 15/16	9¼	4¼	6¾	7⅛	9⅝	2¼	4½	1
4	13	9	1⅛	22	12	1⅛	5 9/16	3¾	10	4⅞	7½	7⅞	10½	2½	5	1⅛
4½	14	9½	1¼	24	12½	1¼	6⅝	4	10⅝	4⅞	7⅝	7⅞	10½	2⅛	5	1⅛
5	15	10½	1¼	25⅞	13	1¼	6¾	4½	11⅝	5⅜	8¼	9⅛	11⅛	3	5	1¼
6	15½	12	1⅜	29⅜	14½	1¼	8	5	13½	6	9½	10⅛	12⅜	3¼	5½	1⅜
7	17	13½	1½	32¾	15½	1¼	8⅜	6¾	14¾	6⅞	10⅞	10⅝	15¼	4½	6	1⅜
8	17½	15	1⅝	36⅛	17	1⅜	9⅜	7⅝	16	7¼	11⅞	11⅛	16⅛	5⅛	6	1½
9	18	16	1¾	39⅜	18½	1⅜	11¼	7½	17½	8⅛	13¼	11⅞	18⅛	5	6⅜	1⅝
10	18	17	1¾	42¼	20	1⅜	11⅞	8¾	18¼	7⅞	14⅜	11⅞	18⅛	6	6⅜	1⅝
12	20	19	2	48⅛	23	1⅜	14	10⅜	20⅜	9½	16¾	13⅜	20⅝	7½	7	1¾
13	20	21	2	52¼	24½	1⅜	17⅜	8¾	22⅜	9⅝	18⅞	14½	23¾	6	8	1⅞
14	20	22	2	55¼	26	1⅜	18¾	9¼	23⅝	9⅞	20¼	14¾	25⅛	6½	9	2

TABLE XXII.

Dimensions of Extension Spindles and Bye-Passes for Fullway Valves (Fig. 42).

A	B	C	D	E	F	G	H	I	J	K	L	M	N	O	P
ins.	ins.	ins.	ins.	ins.	ins.	ins.	ins.	ins.	ins.	ins.	ins.	ins.	ins.	ins.	ins.
2	$\frac{7}{8}$	5					$2\frac{1}{8}$	$5\frac{3}{8}$	$1\frac{1}{2}$	1	$11\frac{7}{8}$				
$2\frac{1}{2}$	$\frac{7}{8}$	5					,,	,,	,,	,,	,,				
3	1	$5\frac{3}{4}$					$2\frac{9}{16}$	6	$2\frac{3}{16}$	,,	$12\frac{1}{2}$				
$3\frac{1}{2}$	1	$5\frac{3}{4}$					,,	7	$2\frac{3}{8}$	$1\frac{1}{8}$	$15\frac{1}{2}$				
4	$1\frac{1}{8}$	$6\frac{1}{2}$					$2\frac{3}{4}$	8	$3\frac{5}{16}$	,,	$16\frac{1}{2}$				
$4\frac{1}{2}$	$1\frac{1}{8}$	$7\frac{1}{8}$					$2\frac{7}{8}$	,,	,,	,,	,,				
5	$1\frac{1}{4}$	$7\frac{5}{8}$					3	$9\frac{3}{8}$	$3\frac{9}{16}$	$1\frac{1}{4}$	$17\frac{7}{8}$				
6	$1\frac{3}{8}$	$8\frac{1}{8}$	9	$6\frac{3}{4}$	$1\frac{3}{8}$	5	$3\frac{1}{4}$	10	$3\frac{5}{8}$	,,	$18\frac{1}{2}$	$3\frac{3}{4}$	5	1	$\frac{1}{2}$
7	$1\frac{3}{8}$	$9\frac{1}{2}$	$9\frac{11}{16}$	,,	,,	,,	$3\frac{3}{8}$	$12\frac{3}{8}$	$3\frac{5}{8}$	,,	$21\frac{3}{8}$	,,	$5\frac{11}{16}$	,,	,,
8	$1\frac{1}{2}$	$10\frac{5}{8}$	$10\frac{13}{16}$	,,	,,	,,	$3\frac{5}{8}$	$14\frac{1}{2}$	$4\frac{1}{16}$	$1\frac{3}{8}$	$23\frac{1}{2}$	,,	$6\frac{3}{16}$	,,	,,
9	$1\frac{5}{8}$	$11\frac{7}{8}$	$10\frac{13}{16}$	$7\frac{7}{8}$	,,	,,	$3\frac{7}{8}$	,,	,,	,,	,,	,,	6	,,	,,
10	$1\frac{5}{8}$	$12\frac{7}{8}$	$11\frac{7}{16}$	,,	,,	,,	,,	,,	,,	,,	,,	,,	7	,,	,,
12	,,	,,	$12\frac{3}{4}$	9	$1\frac{5}{8}$	$5\frac{3}{4}$	,,	,,	,,	,,	,,	,,	$8\frac{7}{16}$	$1\frac{1}{4}$	,,
13	,,	,,	$13\frac{3}{4}$	,,	,,	—	,,	,,	,,	,,	,,	,,	$9\frac{3}{4}$	,,	,,
14	,,	,,	$14\frac{3}{4}$	,,	,,	,,	,,	,,	,,	,,	,,	,,	$10\frac{3}{4}$	,,	,,

Isolating Valves.

Isolating valves made with a heavy ball so arranged as to roll back to a seat in event of a rush of steam towards the boiler to be isolated, are probably dangerous owing to the momentum acquired by the heavy ball under the pressure of the steam. Some-

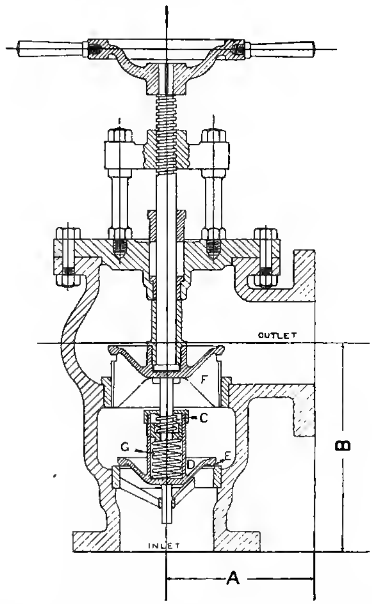


FIG. 43.—ISOLATING VALVE (TEMPLER & RANOE).

thing lighter is more desirable, and in Fig. 43 is shown a combined stop and isolating valve so arranged that when the upper valve *F* is open the lower valve *D* is raised about $\frac{3}{32}$-inch of its seat by the short spring *C*. When steam is flowing the valve rises and compresses the long spring *G*. If the boiler pressure drops below that in the pipes

the difference of a pound per square inch will bring the lower valve to less than $\frac{1}{8}$-inch from its seat, and any further reduction soon closes it. By this valve a boiler at once ceases to be fed from other boilers, and should the fires of any boiler get into bad order at a period of demand for steam, this particular boiler does not become a drag on the others. The isolating valve quite safeguards men inside a boiler, for even if the valve be "opened" the lower valve remains closed, and can only open when full steam-pipe pressure again comes below it.

Exhaust Valves.

The valves in the pipes connecting each engine to a common exhaust main should always be of the fullway type. No bye-pass is necessary. The spindle glands should have a deep box for stuffing, which should be of a soft fibrous order, well lubricated with wax. These valves in exhaust pipes are doubtless a prolific source of air inlets when badly attended to, and a frequent cause of poor vacuum. As they are always cool a fibrous packing comes to no harm.

It need hardly be said that though valve bodies are frequently made of cast iron even for high pressures, and that such castings when used by a reputable firm are apt to be much stronger and sounder than the ordinary run of cast-iron pipe work, still it is safer to employ valves with bodies of cast steel, especially where undue stresses can be foreseen.

Very large valves are made of the equilibrium type, and require no equalizing bye-passes, but they have always proved most difficult to maintain tight because it is practically impossible to screw down two valves rigidly upon two rigid seatings owing to differences of expansion. Some relief might perhaps be secured by a spring device on that valve which is helped to its seat by the steam pressure. A remedy has been attempted by making

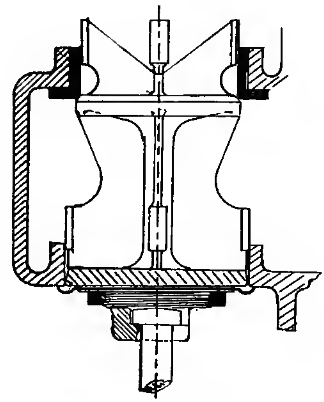


FIG. 44.—FLEXIBLE SEAT VALVE.

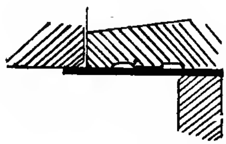


FIG. 45.—FINAL FORM OF FLEXIBLE SEAT OF VALVE.

one of the valves merely a balance piston. Full information on the subject may be found in a paper read to the North-East Coast Institution of Engineers by Mr. J. H. Gibson on December 12, 1902, vol. xix., in which he described the troubles incidental to double-beat valves and the stresses to be dealt with, and finally the experiments with flexible discs which resulted in the adoption of the valve shown in Fig. 44, one valve only in a double-beat

valve being fitted with the disc, the other seating solidly. The disc provides for all distortion and spindle expansion, which combined, cause the trouble in this type of valve. In Fig. 45 is shown the plain flat disc finally adopted in place of the form shown in Fig. 44. It is not possible in the present work to go more fully into the question of valves, which are however a sufficiently important part of a steam pipe system to demand more attention than engineers usually give to them.

To effect the same end as the flexible seat, Holden & Brooke have brought out a double valve in which the two valves are not attached to the same spindle. Each valve has its own independent spindle. These pass through stuffing boxes at opposite ends of the casing, and are pinned to levers with their fulcra on short pivoted links pinned on to the covers. The long end of each lever projects clear of the valve body and carries a nut, in which works a right and left screwed spindle carrying the hand wheel. By this contrivance each valve is pressed upon its seating with equal force, and there is no possibility of one valve leaking while the other is tight by reason of any differential expansion, such as happens with large double-seated valves of common type.

Attention should be directed to the weak feature of all globe valves as Fig. 38, namely, the obstruction they offer to the free flow of water along a pipe. The valve should shut against the pressure where safe to allow this, and there should be a drain from

the base of the globe body steam flowing from the left. Often, as in a balancing header, steam may flow either way, and two separate drains would be needed. A well designed fullway or gate valve is preferable. The larger the valve the more are the faults of globe valves intensified and dangerous.

CHAPTER XII

Drainage

DRAINAGE has already been incidentally referred to in other chapters.

It is an essential provision which is minimized in a good design, and a superfluity of drainage devices indicates faulty design: thus a stop valve at the bottom of a vertical pipe, as when the boiler outlet valve is not at the highest point of a steam range, causes water to collect above the valve, and this must be drained away. If not drained away, then upon opening the steam outlet valve the body of water above the valve may be shot forward like a projectile to rupture the pipe at the first obstruction or square end.

That more accidents have not happened is due to the fact that a boiler joined up to other working boilers has often its steam valve "open" before the boiler can raise the valve. The valve only rises as the pressure on both sides of it become slowly equal, and the collected water falls quietly through the open valve. The steam pipe ought to fall gently all the way to the engine, or to some other drainage point.

Automatic steam traps are connected to such drainage point which remove the water as this collects. Under the term drainage is also understood all those little but important provisions for giving a free flow to water, such as the little bridging pipes across the lower parts of expansion bands when these stand vertically. There must be drainage to the water separator next the engine, and at any point where water can collect. In the best design there will be one drain, viz. that at the water separator only.

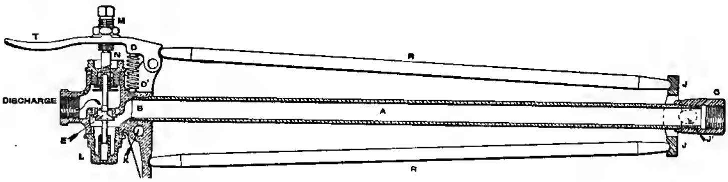


FIG. 46.—EXPANSION STEAM TRAP (HOLDEN & BROOKE).

Steam traps are very various in make, depending some on floats, some on differential expansion. The one example illustrated, for this is a book on pipes rather than accessories, is the expansion trap of Holden & Brooke (Figs. 46–49).

The object of a steam trap is to let water escape from a pipe without letting out any steam. In a form of low-pressure trap a valve is screwed to and from its seat by the rise and fall of a ball float. A small leak allows the ball gradually to fill and sink. It opens a small valve when it sinks, and this admits water from the pipe to be drained. This water further fills up the ball, and

overflows by an inverted syphon from the upper side of the ball. The following steam blows out the whole of the water from the pipe, and finally blows out that in the ball itself, which promptly floats up and closes the escape valve until such time as the small leak into the ball sinks it again and re-establishes the cycle.

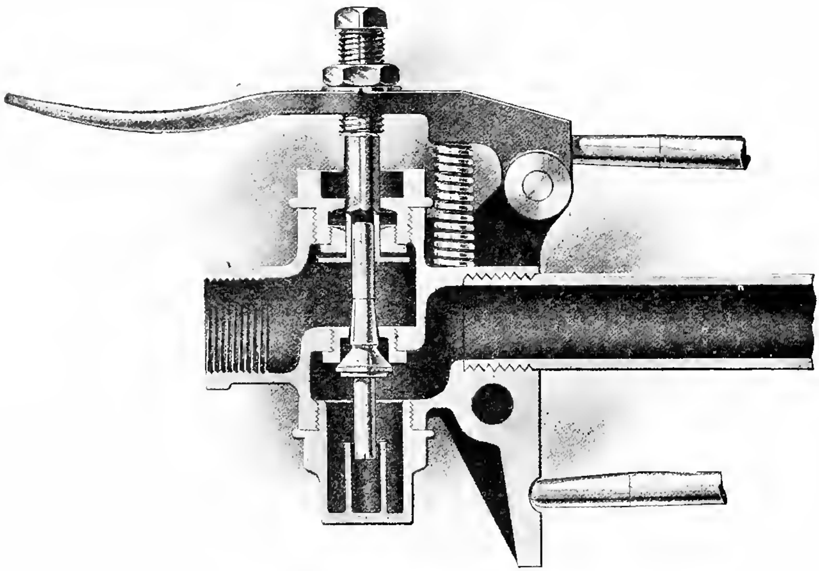


FIG. 47.—EXPANSION STEAM TRAP.

Other taps act by differential expansion as that of Holden and Brooke (Figs. 46–47), shown in more detail in Figs. 48–49.

When the central pipe *A* fills with water and becomes cooler it shortens and pulls the two abutments *DK* against the round-ended strut pieces *R R*, thus opening the valve *E* and compressing the spring

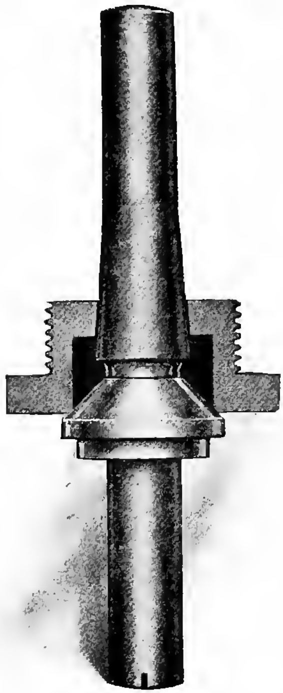


FIG. 48.—VALVE OF STEAM TRAP.

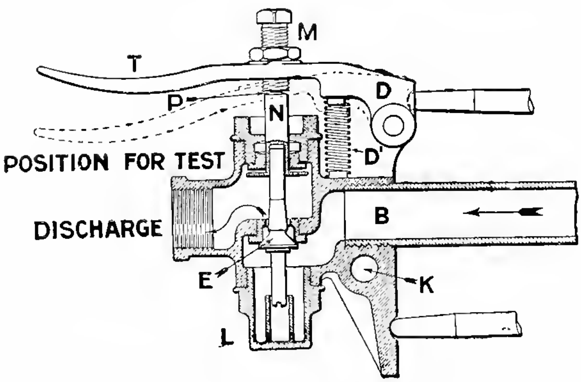


FIG. 49.—SECTION OF EXPANSION STEAM TRAP.

at *D*, which acts promptly to close the valve again as soon as the central pipe expands on becoming again full of steam as the water is expelled. Steam enters at *C* and discharge takes place at the opposite end. The trap may be made to blow at any time by depressing the handle *T*. It is adjusted by the nut *M*.

Steam traps are necessary on drain pipes from the boiler stop-valve where this is so placed as to allow of an accumulation of water above it. A trapped drain must always be placed at the low point of the steam-pipe system which is preferably to be also the steam or water separator.

As traps discharge into the atmosphere the escape from a pipe under pressure is always hotter than steam at atmospheric pressure, and much steam also rises from a trap discharge in consequence of the flashing into steam of a part of the water. The discharge of a trap may go to the pump well, being pure hot water.

CHAPTER XIII

Junction Pieces and Flanges

JUNCTION pieces, such as tees, Y-junctions, crosses, pockets and bends, are often made of cast iron for even high pressures, though many engineers do not approve of this practice, and this subject is elsewhere referred to.

Whatever the material, it is essential that junction pieces should be proportioned on well defined rules. Thus every branch from a given size of T must set the same distance from the face of the branch flange to the centre line of the T, no matter what diameter the branch may be. Similarly this dimension of the T must be the radius of the quarter-bends of the size of the T.

Only by attention to such standard dimensions can a system of pipes be conveniently designed.

Thus a four-way cross piece is simply the same length as a T on each of its pair of faces, and consequently the branch of a T is half the length of the T, while the elbow is like two adjacent flanges of a cross-piece joined by a curve, and the height of a Y-piece is the same as the length over a +, and the spread of the two arms of the Y is equal to the height.

In the Figures 1 to 7 the dimension A is the same for all junctions of the same size, and the four pieces are all dimensioned A or $A \div 2$, and these dimensions as made by the Babcock Co. are given in the annexed table.

TABLE OF TEES, CROSSES, ELBOWS, Y-PIECES, ETC.

	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.
Dia.	2	$2\frac{1}{2}$	3	$3\frac{1}{2}$	4	5	6	7	8	9	10	11	12	14
A=	10	12	12	14	14	16	18	20	21	24	26	28	28	30

The weight of steel pipe with wrought steel flanges as made by the Babcock Co. is given in the Table XIA., p. 37, and the weights of cast equal tees, elbows, Y-pieces and crosses, are as per Table XXIII.

To preserve homogeneity of design wrought junction pieces or cast-steel pieces should have the same linear dimensions as cast pieces.

Under the head of materials will be found other standard dimensions of junction pieces, both cast and wrought, with further remarks on the subject, especially as regards elbows. An elbow strictly must be of the same size $\frac{A}{2}$ as the set of a tee, in order that a pipe system may be homogeneous in design, but this does not apply to true bends which may be of very large radius ; no bend should have a radius less than three diameters, five diameters may be considered a minimum for really first-class practice.

Junction pieces may be, as stated, either of cast iron for limited pressures, cast steel or mild steel for

TABLE XXIII.

WEIGHTS OF JUNCTION PIECES AS PER FIGS. 1–7

Diameter.	2 in.	2½ in.	3 in.	3½ in.	4 in.	5 in.	6 in.	7 in.	8 in.	9 in.	10 in.	11 in.	12 in.	14 in.
T = Lb. . . .	34	62	67	84	103	141	188	263	276	392	498	602	718	915
,, = K. . . .	16	28	31	38	47	64	85	119	129	176	224	271	323	412
Y = Lb. . . .	43	66	83	107	130	181	252	304	320	431	589	750	840	960
,, = K. . . .	20	30	38	49	59	81	114	137	144	194	265	338	378	432
Cross = Lb. . . .	44	64	85	100	124	150	229	336	371	518	661	798	938	1159
,, = K. . . .	20	29	39	45	56	68	103	151	167	233	298	359	422	522
Elbow = Lb. . . .	25	35	47	52	66	90	130	158	228	294	353	404	480	548
,, = K. . . .	12	16	21	24	30	41	59	71	103	133	159	182	216	247

high pressures. The mild steel and the cast steel tee junction, as arranged by Yates & Thom, are

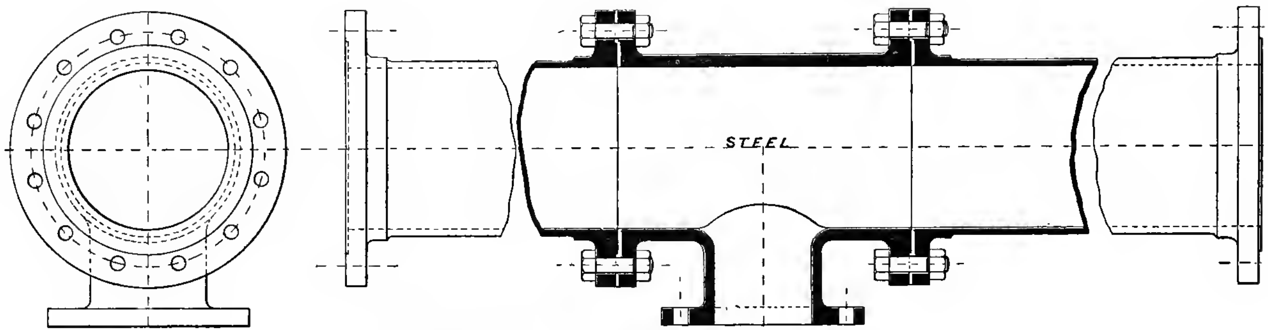


FIG. 50.—CAST JUNCTION PIECE (YATES & THOM).

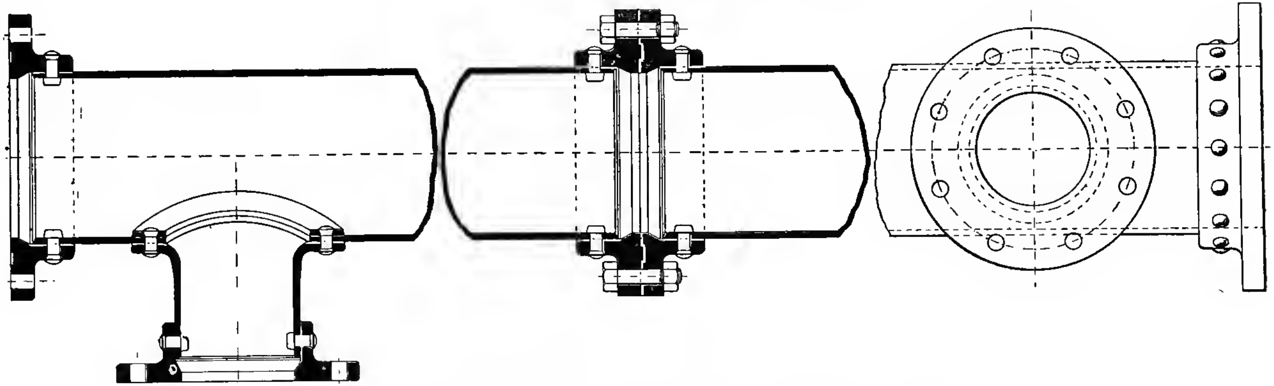


FIG. 51.—MILD STEEL BRANCH CONNECTION (YATES & THOM).

shown in Figs. 50 and 51. The flanges are here shown with the shallow spigot and socket elsewhere described. It is obvious that the recess offers a great safeguard against rupture of the packing, but it is often a great hindrance to the removal of a pipe. It has been suggested that the depth of the recess should be less than the thickness of the joint ring, so that this can be sawn through if necessary to part the pipe. Purely metallic joints of course do not adhere.

Flanges.

Since flanged joints are the most usual, it is of importance that their dimensions should be standardized. The lack of a standard has proved an immense inconvenience. There are five points to be standardized.

(*a*) Flange diameter.

(*b*) Bolt circle diameter.

(*c*) Number of bolts.

(*d*) Size of bolts.

(*e*) Thickness of flange.

(*f*) Angle of bolt holes.

The dimension (*e*) is only for convenience in ordering bolts.

It would be impossible in the space of this book to publish all the principal flange tables.

I have selected a very few only, viz. :—

Those of the Babcock & Wilcox Co., because so largely employed.

The standards of the British Electric Traction Co.,

kindly supplied me by Mr. A. J. Lawson, of that Company, and practically the standard of the Brush Company (Tables XXIV., XXV.; Figs. 52, 53).

The American Standard of 1902.

The German standard is omitted because it employs numbers of bolts not divisible by four, and therefore awkward and academic.

James Russell & Sons' standard.

The cast-iron standard of the Crane Co. of Chicago.

TABLE XXIV.

B.E.T. Co. Standard Steam, Feed Delivery and Suction Pipe Flanges.

Dimensions of Pipes and Flanges.								Bolts.		
A	B	C	D	E	F	G	H	No.	Dia.	Length
in.	in.	in.	in.	in.	in.	in.	in.		in.	in.
$\frac{1}{2}$	$2\frac{1}{2}$	$3\frac{3}{4}$	$\frac{5}{8}$	$\frac{1}{2}$	$\frac{3}{4}$	$1\frac{3}{16}$	$1\frac{5}{16}$	4	$\frac{1}{2}$	$1\frac{5}{8}$
$\frac{3}{4}$	$2\frac{7}{8}$	$4\frac{1}{8}$	$\frac{5}{8}$	$\frac{1}{2}$	$\frac{7}{8}$	$1\frac{1}{2}$	$1\frac{5}{8}$	4	$\frac{1}{2}$	$1\frac{5}{8}$
1	$3\frac{1}{4}$	$4\frac{1}{2}$	$\frac{5}{8}$	$\frac{1}{2}$	$\frac{7}{8}$	$1\frac{7}{8}$	2	4	$\frac{1}{2}$	$1\frac{5}{8}$
$1\frac{1}{4}$	$3\frac{1}{2}$	$4\frac{3}{4}$	$\frac{5}{8}$	$\frac{5}{8}$	1	$2\frac{1}{4}$	$2\frac{3}{8}$	4	$\frac{1}{2}$	$1\frac{7}{8}$
$1\frac{1}{2}$	4	$5\frac{1}{4}$	$\frac{3}{4}$	$\frac{3}{4}$	$1\frac{1}{8}$	$2\frac{1}{2}$	$2\frac{5}{8}$	4	$\frac{5}{8}$	$2\frac{1}{4}$
$1\frac{3}{4}$	$4\frac{1}{2}$	6	$\frac{3}{4}$	$\frac{3}{4}$	$1\frac{1}{4}$	$2\frac{3}{4}$	$2\frac{7}{8}$	4	$\frac{5}{8}$	$2\frac{1}{4}$
2	5	$6\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$1\frac{1}{4}$	$3\frac{1}{8}$	$3\frac{1}{4}$	8	$\frac{1}{2}$	$2\frac{1}{8}$
$2\frac{1}{4}$	5	$6\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$1\frac{1}{4}$	$3\frac{3}{8}$	$3\frac{1}{2}$	8	$\frac{1}{2}$	$2\frac{1}{8}$
$2\frac{1}{2}$	$5\frac{1}{2}$	7	$\frac{5}{8}$	$\frac{3}{4}$	$1\frac{1}{4}$	$3\frac{7}{8}$	4	8	$\frac{1}{2}$	$2\frac{1}{8}$
$2\frac{3}{4}$	$5\frac{3}{4}$	$7\frac{1}{4}$	$\frac{5}{8}$	$\frac{7}{8}$	$1\frac{3}{8}$	4	$4\frac{1}{8}$	8	$\frac{1}{2}$	$2\frac{3}{8}$
3	$6\frac{1}{8}$	$7\frac{1}{2}$	$\frac{5}{8}$	$\frac{7}{8}$	$1\frac{1}{2}$	$4\frac{3}{8}$	$4\frac{1}{2}$	8	$\frac{1}{2}$	$2\frac{3}{8}$
$3\frac{1}{4}$	$6\frac{3}{8}$	$7\frac{3}{4}$	$\frac{3}{4}$	$\frac{7}{8}$	$1\frac{1}{2}$	$4\frac{5}{8}$	$4\frac{3}{4}$	8	$\frac{5}{8}$	$2\frac{1}{2}$
$3\frac{1}{2}$	$6\frac{5}{8}$	8	$\frac{3}{4}$	$\frac{7}{8}$	$1\frac{5}{8}$	$4\frac{7}{8}$	5	8	$\frac{5}{8}$	$2\frac{1}{2}$
$3\frac{3}{4}$	7	$8\frac{1}{2}$	$\frac{3}{4}$	$\frac{7}{8}$	$1\frac{5}{8}$	$5\frac{1}{4}$	$5\frac{3}{8}$	8	$\frac{5}{8}$	$2\frac{1}{2}$
4	$7\frac{1}{2}$	9	$\frac{3}{4}$	1	$1\frac{3}{4}$	$5\frac{1}{2}$	$5\frac{3}{4}$	8	$\frac{5}{8}$	$2\frac{3}{4}$
5	$8\frac{7}{8}$	$10\frac{3}{4}$	$\frac{7}{8}$	$\frac{3}{4}$	2	$6\frac{1}{2}$	$6\frac{3}{4}$	8	$\frac{3}{4}$	$2\frac{3}{8}$
6	10	12	1	$\frac{3}{4}$	2	$7\frac{1}{2}$	$7\frac{3}{4}$	8	$\frac{7}{8}$	$2\frac{1}{2}$
7	11	13	$\frac{7}{8}$	$\frac{3}{4}$	2	$8\frac{1}{2}$	$8\frac{3}{4}$	12	$\frac{3}{4}$	$2\frac{3}{8}$
8	12	14	$\frac{7}{8}$	$1\frac{1}{8}$	$2\frac{1}{4}$	$9\frac{1}{2}$	$9\frac{3}{4}$	12	$\frac{3}{4}$	$3\frac{1}{8}$

TABLE XXV.

B.E.T. Co. Standard Exhaust and Circulating Pipe Flanges.

Dimensions of Pipes and Flanges.									Bolts.		
A	B		C		D	E	F	G	No.	Dia.	Length
in.	ft.	in.	ft.	in.	in.	in.	in.	in.		in.	in.
$2\frac{1}{2}$	0	$5\frac{1}{2}$	0	7	$\frac{5}{8}$	$\frac{5}{8}$	$\frac{3}{8}$	$\frac{1}{2}$	4	$\frac{1}{2}$	2
3	0	6	0	$7\frac{1}{2}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{7}{16}$	$\frac{1}{2}$	4	$\frac{5}{8}$	$2\frac{3}{8}$
$3\frac{1}{2}$	0	$6\frac{1}{2}$	0	8	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{7}{16}$	$\frac{1}{2}$	4	$\frac{5}{8}$	$2\frac{3}{8}$
4	0	$7\frac{1}{2}$	0	9	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{1}{2}$	$\frac{5}{8}$	8	$\frac{1}{2}$	$2\frac{1}{4}$
$4\frac{1}{2}$	0	8	0	$9\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{1}{2}$	$\frac{5}{8}$	8	$\frac{1}{2}$	$2\frac{1}{4}$
5	0	$8\frac{3}{4}$	0	$10\frac{1}{2}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{1}{2}$	$\frac{5}{8}$	8	$\frac{5}{8}$	$2\frac{3}{8}$
6	0	10	1	0	$\frac{3}{4}$	$\frac{7}{8}$	$\frac{1}{2}$	$\frac{5}{8}$	8	$\frac{5}{8}$	$2\frac{5}{8}$
7	0	$10\frac{3}{4}$	1	1	$\frac{3}{4}$	$\frac{7}{8}$	$\frac{9}{16}$	$\frac{5}{8}$	8	$\frac{5}{8}$	$2\frac{5}{8}$
8	1	0	1	2	$\frac{3}{4}$	$\frac{7}{8}$	$\frac{9}{16}$	$\frac{3}{4}$	8	$\frac{5}{8}$	$2\frac{5}{8}$
9	1	1	1	3	$\frac{5}{8}$	1	$\frac{9}{16}$	$\frac{3}{4}$	12	$\frac{1}{2}$	3
10	1	$2\frac{1}{4}$	1	$4\frac{1}{4}$	$\frac{3}{4}$	1	$\frac{5}{8}$	$\frac{3}{4}$	12	$\frac{5}{8}$	3
11	1	3	1	5	$\frac{3}{4}$	1	$\frac{5}{8}$	$\frac{3}{4}$	12	$\frac{5}{8}$	3
12	1	5	1	$7\frac{1}{4}$	$\frac{7}{8}$	$1\frac{1}{8}$	$\frac{3}{4}$	$\frac{3}{4}$	12	$\frac{3}{4}$	$3\frac{3}{8}$
13	1	6	1	$8\frac{1}{2}$	$\frac{7}{8}$	$1\frac{1}{4}$	$\frac{3}{4}$	$\frac{7}{8}$	12	$\frac{3}{4}$	$3\frac{3}{4}$
14	1	7	1	$9\frac{1}{2}$	$\frac{7}{8}$	$1\frac{1}{4}$	$\frac{3}{4}$	$\frac{7}{8}$	12	$\frac{3}{4}$	$3\frac{3}{4}$
16	1	$9\frac{1}{2}$	2	0	1	$1\frac{3}{8}$	$\frac{7}{8}$	$\frac{7}{8}$	12	$\frac{7}{8}$	$3\frac{7}{8}$

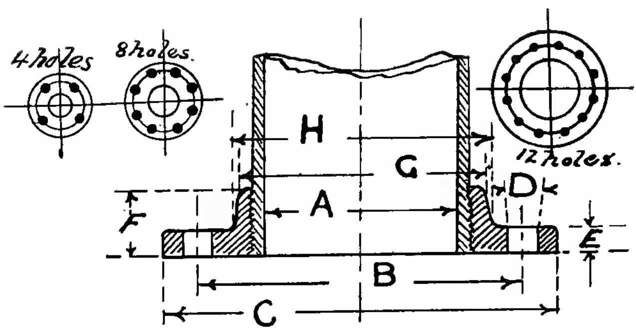


FIG. 52.—FLANGES (B.E.T. CO. STANDARD).

As regards the number of bolts this should always be divisible by four, and should never be less than eight, if eight bolts can be got in. The size of a bolt should not be less than $\frac{5}{8}$-inch, and the holes should be $\frac{1}{8}$-inch larger than the bolts. This rule gives excessive bolt strength in some sizes where the jump is made to an additional four bolts, but the $\frac{1}{2}$-inch bolt is not a satisfactory thing in practical engineering unless of manganese steel.

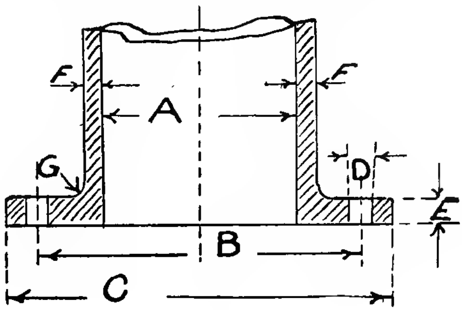


FIG. 53.

The advantage of the multiple of four is that a piece can always be turned through an angle of 90° and bolts will still come right. The arrangement of bolt holes should always be so that no hole comes on a centre line.

The pitch of bolts should not exceed $4\frac{1}{2}$ inches, according to Mr. Atkinson. As soon as with a given number of bolts the pitch becomes $4\frac{1}{2}$ inches, the next size of pipe should have an additional four bolts. Thus with his 7-inch pipe the bolt circle is $11\frac{1}{2}$ inches, and the pitch of 8 bolts is 4·51. His 8-inch pipe, therefore, has 12 bolts.

TABLE XXVI.

Babcock & Wilcox Standard Flanges.

Bore.	High Pressure Steam. Diam. of Flanges	No. of Bolts.	Diam. of Pitch Circle.	Size of Bolts.	Exhaust. Diam. of Flanges	No. of Bolts.	Diam. of Pitch Circle.	Size of Bolts.	Feed. Diam. of Flanges	No. of Bolts.	Diam. of Pitch Circle.	Size of Bolts.
ins.	ins.		ins.	in.	ins.		ins.	in.	ins.		ins.	in.
¾	3½	4	2½	⅜	—	—	—	—	3½	4	2½	⅜
1	4½	4	3¼	½	—	—	—	—	4½	4	3¼	½
1¼	4½	4	3½	½	—	—	—	—	4½	4	3½	½
1½	5	4	4	½	—	—	—	—	5	4	4	½
2	7	6	5½	⅝	—	—	—	—	6	4	4½	⅝
2½	7¾	6	6	⅝	—	—	—	—	7	4	5½	⅝
3	8¾	6	6¾	¾	—	—	—	—	8¼	4	6½	¾
3½	9	6	7¼	¾	—	—	—	—	—	—	—	—
4	10	8	8¼	¾	9	4	7¼	¾	9½	6	7¾	¾
5	11	8	9	¾	10¼	6	8½	¾	—	—	—	—
6	12	8	10	¾	11½	6	9¾	¾	—	—	—	—
7	14	12	11¾	¾	12½	6	10¾	¾	—	—	—	—
8	14	12	12¼	¾	13¾	8	12	¾	—	—	—	—
9	15	12	13¼	¾	14¾	8	13	¾	—	—	—	—
10	17	12	14¾	⅞	15¾	8	14	¾	—	—	—	—
11	18¼	12	16	⅞	16¾	12	15	¾	—	—	—	—
12	19½	12	17¼	⅞	17¾	12	16	¾	—	—	—	—
13	—	—	—	—	19¼	12	17½	¾	—	—	—	—
14	—	—	—	—	20¼	12	18½	¾	—	—	—	—
15	—	—	—	—	21¼	12	19½	¾	—	—	—	—
16	—	—	—	—	22¼	12	20½	¾	—	—	—	—
17	—	—	—	—	23½	16	21¾	¾	—	—	—	—
18	—	—	—	—	24½	16	22¾	¾	—	—	—	—
20	—	—	—	—	26½	16	24¾	¾	—	—	—	—
22	—	—	—	—	28¾	20	27	¾	—	—	—	—
24	—	—	—	—	30¾	20	29	¾	—	—	—	—

The author is inclined rather to limit the pitch to 4 inches, thus giving 12 bolts to the 7-inch pipe as practised by the Crane Co.

Ordinary commercial bolts tested by Professor Goodman have shown a tensile strength of 29 to 35 tons per square inch, the smaller bolts coming

out best, but bolts in practice are exposed to a torsional stress, and the smaller bolts are apt to have the biggest stresses put on them, and it is wise to keep to $\frac{5}{8}$ as a minimum size where possible.

TABLE XXVII.

DIMENSIONS OF CAST-IRON FLANGED FITTINGS AND CONNECTIONS, AS USED BY CRANE COMPANY, CHICAGO.

Inside Diameter of Pipe.	Diameter of Flange.	Diameter of Bolt Circle.	Number of Bolts.	Set of T-Branch or Quarter Bend, etc.	Length of a T.
ins.	ins.	ins.		ins.	ins.
2	6	$4\frac{3}{4}$	4	$4\frac{1}{2}$	9
$2\frac{1}{2}$	7	$5\frac{1}{2}$	4	5	10
3	$7\frac{1}{2}$	6	4	$5\frac{1}{2}$	11
$3\frac{1}{2}$	$8\frac{1}{2}$	$6\frac{3}{4}$	4	6	12
4	9	$7\frac{1}{4}$	8	$6\frac{1}{2}$	13
$4\frac{1}{2}$	$9\frac{1}{4}$	$7\frac{3}{4}$	8	7	14
5	10	$8\frac{1}{4}$	8	$7\frac{1}{2}$	15
6	11	$9\frac{1}{4}$	8	8	16
7	$12\frac{1}{2}$	11	12	$8\frac{1}{2}$	17
8	$13\frac{1}{2}$	12	12	$9\frac{1}{2}$	19
9	15	13	12	$10\frac{3}{4}$	$21\frac{1}{2}$
10	16	$14\frac{1}{4}$	12	$11\frac{1}{2}$	23
12	19	17	16	$12\frac{3}{4}$	$25\frac{1}{2}$
14	21	$18\frac{1}{2}$	16	$13\frac{1}{4}$	$26\frac{1}{2}$
16	$23\frac{1}{2}$	$21\frac{1}{2}$	20	$15\frac{1}{4}$	$30\frac{1}{2}$
18	25	$22\frac{1}{2}$	20	$16\frac{1}{2}$	33
20	$27\frac{1}{2}$	$24\frac{3}{4}$	20	18	36
22	$29\frac{1}{2}$	$27\frac{1}{2}$	24	20	40
24	$31\frac{1}{2}$	$29\frac{1}{2}$	24	22	44

For very special work bolts of manganese steel may be procured, which are greatly superior to ordinary bolts.

Mr. E. R. Briggs proposes as a suitable stress for bolts, $f = 5{,}000\, d$, where d is the nominal bolt dia-

meter. This gives the following stresses per square inch to be allowed in any bolt :—

Bolt.		*f*
$\frac{1}{2}$-in.	=	2,500 pounds
$\frac{5}{8}$,,	=	3,125 ,,
$\frac{3}{4}$,,	=	3,750 ,,
$\frac{7}{8}$,,	=	4,375 ,,
1 ,,	=	5,000 ,,
$1\frac{1}{8}$,, and over	=	5,625 ,,

He would never allow *f* to exceed 6,000 pounds. The rule allows for the weakness and liability to overstress of small bolts.

TABLE XXVIII.

STANDARD FLANGES ADOPTED BY MESSRS. JAMES RUSSELL AND SONS, CROWN TUBE WORKS, WEDNESBURY.

Inside Diameter of Pipe.	Outside Diameter of Flange.	Inside Diameter of Pipe.	Outside Diameter of Flange.
inches.	inches.	inches.	inches.
$\frac{1}{2}$	$3\frac{1}{2}$	5	10
$\frac{3}{4}$	$3\frac{3}{4}$	$5\frac{1}{2}$	$10\frac{1}{2}$
1	$4\frac{1}{2}$	6	$11\frac{1}{2}$
$1\frac{1}{4}$	5	$6\frac{1}{2}$	$12\frac{1}{2}$
$1\frac{1}{2}$	$5\frac{1}{2}$	7	13
$1\frac{3}{4}$	$5\frac{1}{2}$	$7\frac{1}{2}$	$13\frac{1}{2}$
2	6	8	14
$2\frac{1}{4}$	$6\frac{1}{2}$	$8\frac{1}{2}$	$14\frac{1}{2}$
$2\frac{1}{2}$	7	9	15
$2\frac{3}{4}$	$7\frac{1}{2}$	$9\frac{1}{2}$	$15\frac{1}{2}$
3	8	10	$16\frac{1}{2}$
$3\frac{1}{4}$	$8\frac{1}{2}$	$10\frac{1}{2}$	17
$3\frac{1}{2}$	$8\frac{1}{2}$	11	$17\frac{1}{2}$
$3\frac{3}{4}$	9	$11\frac{1}{2}$	18
4	9	12	$18\frac{1}{2}$
$4\frac{1}{2}$	$9\frac{1}{2}$		

TABLE XXIX.

STANDARD FLANGES ADOPTED IN AMERICA, JANUARY, 1, 1902, FOR PRESSURES 101 LB. TO 250 LB.

Diameter of Pipe.	Diameter of Flange.	Thickness of Flange.	Diameter of Bolt Circle.	Number of Bolts.	Size of Bolts.
ins.	ins.	ins.	ins.		ins.
2	6½	⅞	5	4	⅝
2½	7½	1	5⅞	4	¾
3	8¼	1⅛	6⅝	8	⅝
3½	9	1 3/16	7¼	8	⅝
4	10	1¼	7⅞	8	¾
4½	10½	1 5/16	8½	8	¾
5	11	1⅝	9¼	8	¾
6	12½	1 7/16	10⅝	12	¾
7	14	1½	11⅞	12	⅞
8	15	1⅝	13	12	⅞
9	16	1¾	14	12	⅞
10	17½	1⅞	16¼	16	⅞
12	20	2	17¾	16	⅞
14	22½	2⅛	20	20	⅞
15	23½	2 3/16	21	20	1
16	25	2¼	22½	20	1
18	27	2⅜	24½	24	1
20	29½	2½	26¾	24	1⅛
22	31½	2⅝	28¾	28	1⅛
24	34	2¾	31¼	28	1⅛

CHAPTER XIV

Separators, Exhaust Heads and Atmospheric Valves

A WATER separator for removing the surplus water from saturated steam acts always on the principle of the first law of motion, taking into effect the tendency of an inert body such as water to move in a straight line. All separators, therefore, act by causing the flow of steam to be suddenly reversed in direction. The steam follows the new path and the water continues, and is caught in a suitable receptacle and trapped off. There are legions of separators in the market, but all work on the same principle. Two or three forms only are therefore illustrated, that of Holden & Brooke (Fig. 54), and those of Yates & Thom (Figs. 55, 56).

EXHAUST HEADS,

for preventing the escape of oil and water at atmospheric discharges, act on the same principle, affording a large internal area for the steam, and a quiet part for the collection of oil gathered in the cones and on the outer cylinder.

Fig. 57 is the exhaust head of Holden & Brooke, who make them so that a 5-inch head will deal with 1,000 pounds of exhaust steam per hour, a 10-inch

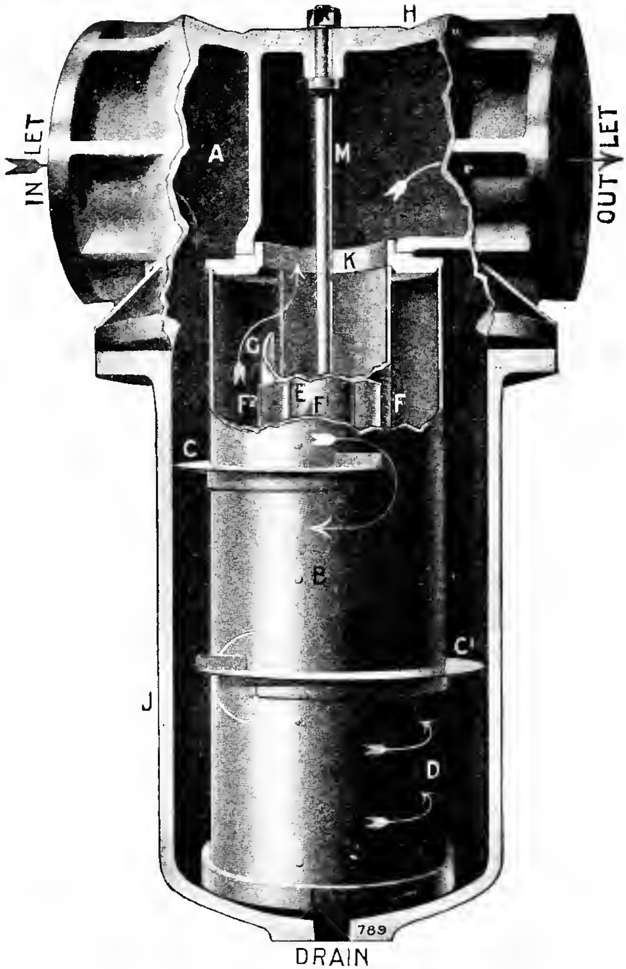


FIG. 54.—WHIRLING SEPARATOR (HOLDEN & BROOKE).

with 3,060 pounds, a 16-inch with 9,000 pounds, and a 24-inch with 25,000 pounds, and *pro rata.*

Atmospheric Valves.

Where an alternate exhaust is desired to a condenser, or to atmosphere, a valve is fitted in the

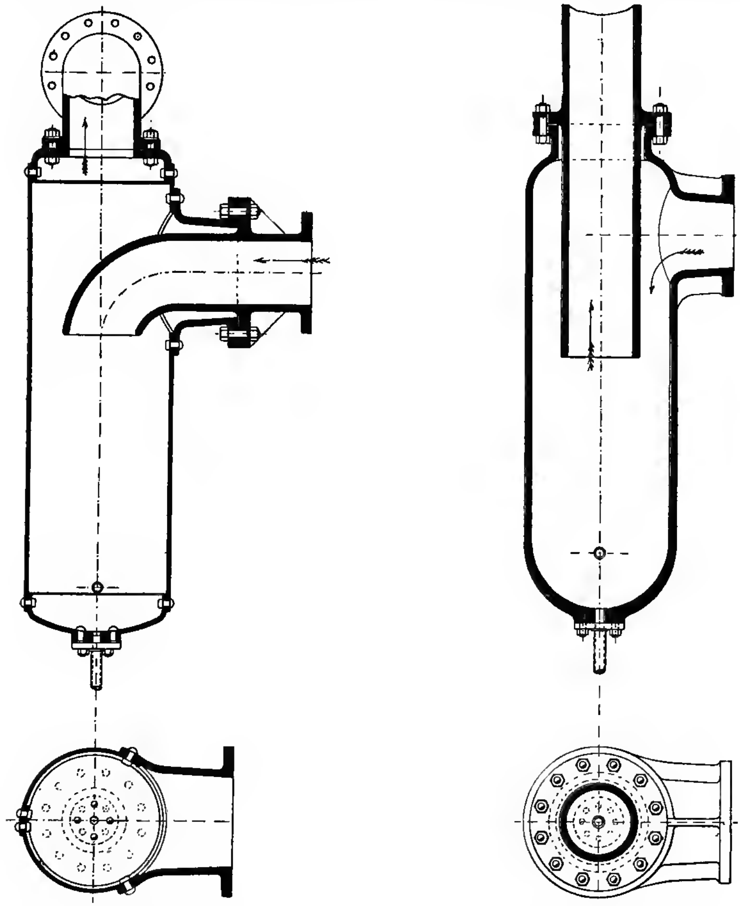


FIG. 55. FIG. 56.

REVERSE FLOW STEAM AND WATER SEPARATORS (YATES & THOM).

atmospheric exhaust proper, which will automatically open and close when condensation ceases or

resumes. These valves ought to have a shallow water seal above them so as to obviate any air leakage. An oil dashpot ought to be fitted outside the body to prevent hammering of the valve, which will occur if no dashpot is present or only an air dashpot be employed.

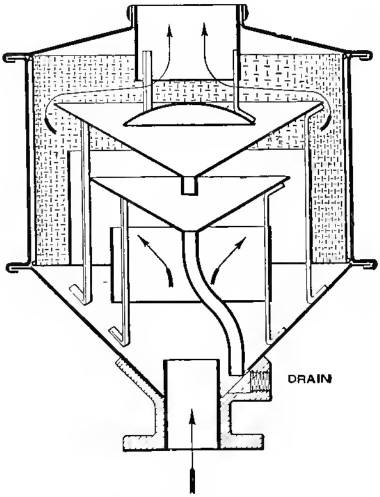


FIG. 57.—EXHAUST HEAD (HOLDEN & BROOKE).

A glass water-gauge should show the depth of water-seal, and a drain pipe should prevent its becoming too deep. A supply pipe should also keep up a supply of water or the seal may leak away and air may leak in. The atmospheric valve is so frequent a cause of bad vacuum that it deserves more attention than it usually obtains. One of these valves by Templer & Ranoe is shown in Fig. 58. It can be placed upside down equally well. There is an outside oil dashpot. An automatic exhaust

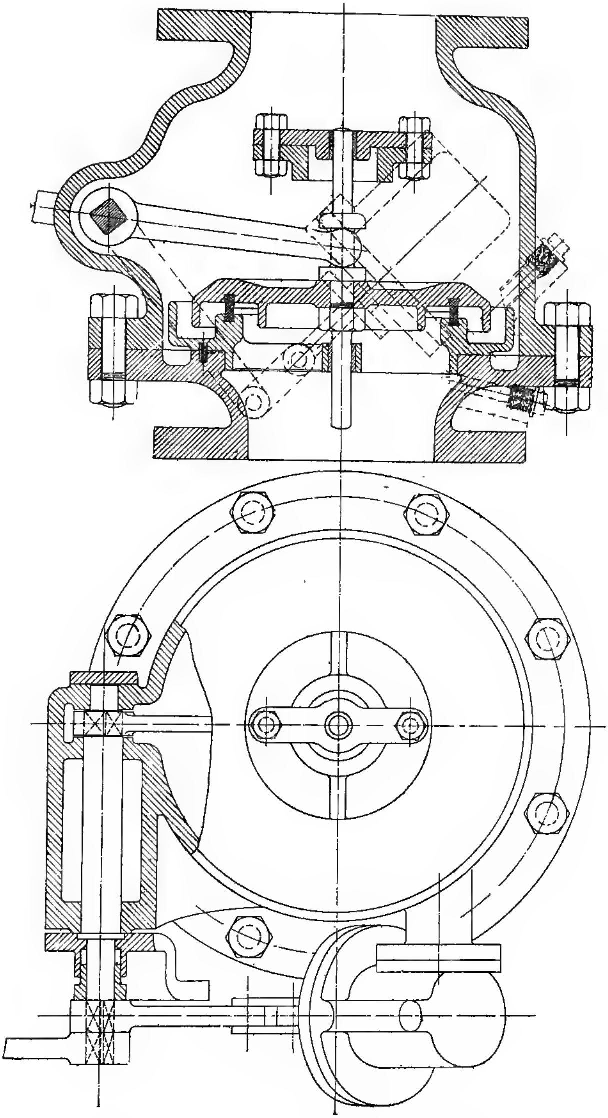


FIG. 58.—ATMOSPHERIC EXHAUST VALVE.

valve by Thos. Walker, of Tewkesbury, is shown in Fig. 59. This is shown with the customary air dashpot, but oil can be substituted. In the author's

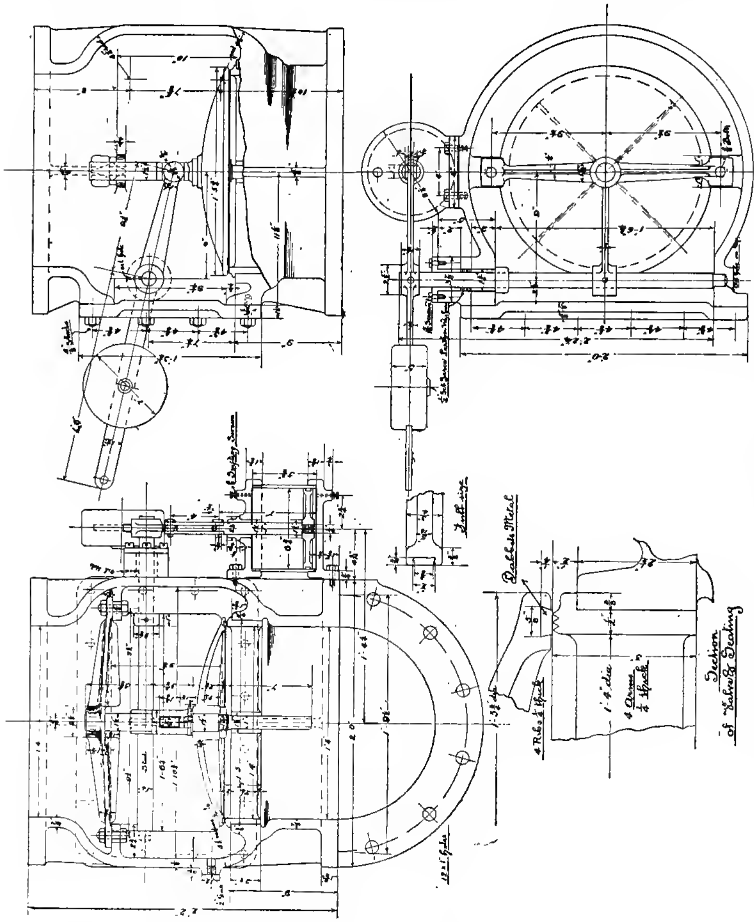


FIG. 59.—ATMOSPHERIC EXHAUST VALVE, BY THOS. WALKER, TEWKESBURY.

opinion the air dashpot is what is responsible for the clatter of atmospheric valves when opening and closing under the pulsation of the exhaust. He would fill the dashpot with oil on both sides of the piston,

and in place of the snifting valves would connect the top and bottom of the cylinder by a small pipe with a valve. By regulating this valve the proper action of the automatic exhaust valve would be better secured. An air dashpot can be converted into an oil dashpot by means of a small cock and a bit of pipe joining the opposite ends of the cylinder.

CHAPTER XV

Superheated Steam

THE fact is well recognized that moisture in steam is one of the great causes of friction and resistance to flow. The water is inert. When it strikes the pipe surface it is stopped in its progress, and it continually puts a drag on the steam. Steam dried and superheated undoubtedly travels faster, but experiment is wanting to say to what extent. It is possible that more superheated steam will pass through a given pipe in a given time with a given loss of pressure than is the case with saturated steam.

The volume of superheated steam varies with its absolute temperature very approximately. Thus steam at 360° F. has an absolute temperature of 360 + 459, or 819°. Superheated 100° F., its volume is now increased in the ratio (819 + 100) : 819, or about 12 per cent. Superheated 200° F., the volumetric increase is nearly 24 per cent. With modern pressures the temperature of superheat will rarely exceed 200° above saturation temperature.

Mr. Cruse makes the cross-section of the pipes of his superheater from 25 per cent. for high pressures to 50 per cent. for low pressures in excess of the

boiler steam pipe. With this provision the loss of pressure in traversing the long pipe superheater is always under three pounds, and more usually only one pound to two pounds. The superior mobility of superheated steam is probably such that the size of the steam pipe need not be increased for a given weight of steam. Where the same power is to be developed, the diminution of the weight of steam required will not differ far from the inverse ratio of the increase in volume due to superheat. On the whole, therefore, for a given power the steam pipes may be less in size than usually provided.

Pipe Coverings.

Every manufacturer of pipe coverings will produce figures to show that his special material is the best.

It is certain that almost anything sold will pay for itself in steam saved.

The best of all material is loose wool. Loose lamp-black, down and hair-felt come next, and generally it may be said that the best heat insulators are those which imprison the most air in a finely divided condition. But all organic substances are unsuitable for the modern conditions with superheated steam, and some form of magnesia covering or other similar preparation is probably best.

Coverings are sometimes put on in a soft plastic condition and hardened in place by the heat of the

pipe. Others are built up into sections and fitted to the pipes and held by wire-binding, or clips of hoop-iron, or by hooks and eyes. The neatest covering has an outer case of Russia iron. In all cases the flanges ought to be covered. No covering should be less than one inch in thickness. This may be exceeded if the value of the heat saved renders it economical, and often it will pay to put two inches of covering upon a pipe.

Numerous tests have been made from time to time by various experimenters on different substances, and particulars of these tests may be found in the *Proceedings of the American Society of Mechanical Engineers*. Tables and data may be found in Kempe's *Year Book* and in *Steam*, and in various other pocket-books and in the catalogues of makers of coverings. From one of these it appears that the composition prevented five-sixths of the loss with bare pipes.

A thickness of one inch of hair-felt also reduced condensation to one-sixth, two inches of felt reduced it to one-eleventh, while the abnormal thickness of six inches reduced it to one-twenty-fourth.

Small pipes lose relatively more than large pipes because the area of an equal thickness of covering is greater. It is also more costly to cover small pipes because the same thickness or more is necessary, and the area of a small pipe and its steam-carrying capacity is less per unit of superficial area. As a general rule it will be good practice to employ coverings $1\frac{1}{2}$ inch thick, unless experiment can

be made to determine the economy of a different thickness by equating the interest charge of the covering and the fuel value of the heat loss, remembering that in a hardly-pressed plant an additional outlay on pipe coverings might render it possible to avoid adding extra boilers.

It is possible of course to cover a pipe first with magnesia, and upon this with hair-felt, which would be protected from the most severe heat by the inner mineral layer.

Coverings which are liable to loosen or disintegrate under vibration should be avoided.

Slag-wool is apt to do this, and it is heavy and is liable to cause trouble if it gets into machinery bearings. Lightness is a favourable quality in a covering because it indicates considerable air space, a feature which is sought in fossil meal in the shape of the minute cavities of the diatoms; in certain asbestos, corrugated millboards, and in wool-felt, both in the fibre itself and the frictional effect by which the myriads of fibres hold the air from circulating. The non-circulation of air inside the mass of the covering is one of the more valuable features of the best compositions.

The very common practice of leaving pipe flanges and bolt heads and nuts bare of protection is necessary with the plastic compounds which are stopped off short of the flanges, but this is no excuse for neglecting to provide loose covers over the flanges.

A few tables and deductions abstracted from a report by Mr. Atkinson, of Boston, relative to the

tests of Mr. C. E. L. Norton, on pipe coverings, will be useful. They have been translated on the basis of £1 = $5, and they are probably as accurate as any tests made, and will afford a useful guide to the engineer who wishes to have his pipes dealt with in the manner that the importance of the question demands.

The tests were made in 1898 by Mr. Norton, on pipe coverings of various types. He employed an electrically-heated apparatus with coils of wire in a bath of oil, and by maintaining the oil at a fixed temperature he was able to measure the heat generated, and therefore lost, by the measure of the current consumed. Particulars of the test need not be detailed; they may be found in Circular Note of the Mutual Boiler Insurance Co., of 31, Milk Street, Boston, U.S.A., 1898.

A few of the tabulated results are here abstracted, and it may be added that Mr. Edward Atkinson, selected A, D, G, E as safe in respect of safety from fire and efficiency in results.

Articles containing lime sulphate are not advised because of the danger of corrosion of the covered pipe, and many so-called magnesia coverings contain rather lime sulphate than magnesia. Magnesia of course is good if it can be obtained really pure and unadulterated. Mr. Norton also recommended plastic coverings as better than sectional for certain conditions, and especially where vibration is likely to occur. Sectional coverings are looked on usually as better than plastic. Yet at least 20 per cent. of

plastic must always be employed for the irregular surfaces.

The tables which follow are at least sufficient as a general guide, and prove the undisputed benefit of good coverings.

Specimen A, Nonpareil cork standard, consists of granulated cork, pressed in a mould at high temperature and then submitted to a fire-proofing process.

Specimen B, Nonpareil cork octagonal, is similar in composition, but is made up of several strips of cork, instead of two semi-cylindrical sections.

Specimen C, Manville high-pressure sectional cover, is composed of an inner jacket of earthy material and an outer jacket of wool-felt, the whole being one and one-quarter inches thick.

Specimen D, magnesia, is a moulded, sectional cover, composed of about 90 per cent. carbonate of magnesia.

Specimen E is essentially an air cell cover, being composed of sheets of asbestos paper which has been indented before being laid up, the indentations serving to keep the thin sheets of paper from coming into close contact with one another, thereby causing a considerable amount of air to be held throughout the body of the cover.

Specimen F is composed of a wool-felt with a lining of asbestos paper.

Specimen G is a cover made up of thin sheets of asbestos paper, fluted or corrugated, and stuck together with silicate of soda.

TABLE A.

Specimen.	Name.	B. Th. U. loss per sq. ft. pipe surface per minute.	Per cent. or ratio of loss to loss from bare pipe.	Thickness in inches.	Weight in ounces per ft. of length, 4 in. diam.
A	Nonpareil Cork Standard	2·20	15·9	1·00	27
B	,, ,, Octagonal	2·38	17·2	·80	16
C	Manville High Pressure	2·38	17·2	1·25	54
D	Magnesia	2·45	17·7	1·12	35
E	Imperial Asbestos	2·49	18·0	1·12	45
F	"W. B."	2·62	18·9	1·12	59
G	Asbestos Air Cell	2·77	20·0	1·12	35
H	Manville Infusorial Earth	2·80	20·2	1·50	—
I	,, Low Pressure	2·87	20·7	1·25	—
J	,, Magnesia Asbestos	2·88	20·8	1·50	65
K	Magnabestos	2·91	21·0	1·12	48
L	Moulded Sectional	3·00	21·7	1·12	41
O	Asbestos Fire Board	3·33	24·1	1·12	35
P	Calcite	3·61	26·1	1·12	66
	Bare Pipe	13·84	100·	—	—

Specimen H is a plastic covering made of infusorial earth.

Specimen I is similar to Specimen F.

Specimen J is a plastic cover called magnesia-asbestos. It contains only a slight amount of carbonate of magnesia.

Specimen K is a moulded cover, containing about 45 per cent. of carbonate of magnesia and a considerable percentage of carbonate of calcium.

Specimen L is composed mainly of sulphate of calcium and some 20 per cent. of $MgCO_3$ and has upon its outer surface a thick sheet of felt board.

Specimen O is similar to Specimen G, except that it has larger cells and contains much more silicate of soda. It is very hard and strong.

Specimen P is a sectional, moulded cover, composed mainly of sulphate of calcium. It has an outer layer of felt board.

Of Specimens C, J, L, and P, the principal ingredient is stated to be sulphate of lime and *not* carbonate of magnesia. Prospective purchasers of pipe covers should not be misled by names. Since the appearance of Professor Ordway's reports it has been recognized that carbonate of magnesia is of great value as a non-conductor of heat, hence the name "magnesia" has been applied to a great many covers. It is to be observed that there is no virtue in a name. Asbestos is merely an incombustible material in which air may be entrapped, but when not porous is a good conductor of heat. Magnesia is a most effective non-conductor. This

name has been applied to many compounds of which the greater part consists of carbonate of lime or of plaster of Paris, materials which are not good as heat retarders. The percentage of magnesia carbonate and plaster of Paris in several moulded, sectiona' covers is given in Table B.

The Cork, Magnesia, Air Cell, and Imperial covers cause no corrosion.

TABLE B.

Specimen.	Percentage Composition.	
	$MgCO_3$ Carbonate of Magnesia.	$CaSO_4$ Sulphate of Calcium.
D	80 to 90	3
C	less than 5	65 to 75
L	20 to 25	50 to 60
P	less than 5	75
J	10 to 15	none

The conditions of testing were reasonably near the conditions of actual practice. The room temperature was kept at 72° F. and the openings into the room were carefully closed. It was found early in the series that variation in the amount of moisture present in the air altered the amount of heat lost from the covers, but no attempt was made to correct this. The error introduced is not greater than 1 per cent.

It was found that the heat loss per square inch of the flat surfaces at the ends of the pipes was less by several per cent. than the loss from the sharply

curved sides, and as all pipe covers tested were used to cover both sides and ends, the figures given in the table show a loss, less than would be shown were the pipe surface wholly cylindrical, and more than if it were all flat.

The pipes were suspended from the ceiling, as described in an early paragraph, and the air circulating about them was due only to their own convection currents. The variation in thickness in different places on the same specimen was considerable, but an average of twenty measurements was taken and results given in the table to the nearest one-eighth of an inch. Owing to these variations in thickness, the results of a measurement of the efficiency of any one cover cannot be used to predict the efficiency of a second cover of the same make with an accuracy greater than 2 per cent. Two specimens of each make were tested, and, in some cases, four, the mean value being given in the table.

Table C gives the saving, in £'s, due to the use of the various covers.

Table D shows that at the end of ten years the best of the covers tested will have saved £9·2 more than the poorest. The difference between the several covers of the better grade is exceedingly small.

The money saving is computed on the following assumptions :—Coal at sixteen shillings a ton evaporates ten pounds of water per pound of coal ; the pipes are kept hot ten hours a day, three hundred and ten days a year. If computations are made, as is sometimes done, on an assumption that the

TABLE C.

Specimen.	Name.	Loss per sq. ft. B. Th. U. at 200 lb.	Saving B. Th. U. per sq. ft.	Saving per year per 100 sq. ft.
	Nonpareil Cork Standard	2·20	11·64	£7·56
B	,, ,, Octagonal	2·38	11·46	7·44
C	Manville Sectional High Pressure	2·38	11·46	7·44
D	Magnesia	2·45	11·39	7·38
E	Imperial Asbestos	2·49	11·35	7·36
F	"W. B."	2·62	11·22	7·28
G	Asbestos Air Cell	2·77	11·07	7·20
H	Manville Infusorial Earth	2·80	11·04	7·17
I	,, Low Pressure	2·87	10·97	7·13
J	,, Magnesia Asbestos	2·88	10·96	7·12
K	Magnabestos	2·91	10·93	7·10
L	Moulded Sectional	3·00	10·84	7·04
O	Asbestos Fire Board	3·33	10·51	6·84
P	Calcite	3·61	10·23	6·65
	Bare Pipe	13·84	—	—

pipes are hot twenty-four hours a day, three hundred and sixty-five days in a year, the saving is nearly three times that shown in Table C.

Generally speaking, a cover saves heat enough to pay for itself in a little less than a year at three hundred and ten ten-hour days, and in about four months at three hundred and sixty-five twenty-four hour days.

It is evident that the decision as to the choice of cover must come from other considerations, as well as from the conductivity.

The question of the ability of a pipe cover to withstand the action of heat for a prolonged period without being destroyed or rendered less efficient is of vital importance. The increasing use of cork as an insulator has led to many questions as to its ability to remain "fire-proof." Exposed to a temperature corresponding to three hundred and fifty pounds of steam for three months, and to a temperature corresponding to one hundred pounds for two years, no change was found, and any suspicion of its ability to withstand continued heating is considered groundless.

The magnesia covering is of course unquestionable on this ground, being almost indestructible by heating.

The Imperial asbestos is also perfectly safe from any fire risks, as is the Air-Cell and Fire-Board.

The Manville infusorial earth, and also the Manville magnesia-asbestos are liable to no accident from fire, nor is the Carey calcite.

TABLE D.

Net Saving per 100 Sq. Ft.

Specimen.	Name.	1 Year.	2 Years.	5 Years.	10 Years.
A	Nonpareil Cork Standard	£2·56	£10·12	£32·80	£70·60
B	" " Octagonal	2·44	9·88	32·20	69·40
C	Manville Sectional High Pressure	2·44	9·88	32·20	69·40
D	Magnesia	2·38	9·76	31·90	68·80
E	Imperial Asbestos	2·36	9·72	31·80	68·60
F	"W. B."	2·28	9·56	31·40	67·80
G	Asbestos Air Cell	2·20	9·40	31·00	67·00
H	Manville Infusorial Earth	2·17	9·34	30·85	66·50
I	" Low Pressure	2·13	9·26	30·75	66·40
J	" Magnesia Asbestos	2·12	9·24	30·60	66·20
K	Magnabestos	2·10	9·20	30·50	66·00
L	Watson's Moulded Sectional	2·04	9·08	30·20	65
O	Asbestos Fire Board	1·84	8·68	29·20	63·40
P	Calcite	1·65	8·30	28·24	61·40
Q	Bare Pipe	—	—	—	—

It is not safe to put upon a steam-pipe wool, hair-felt, or woollen felt in any form. The causes of risk are two: First, the wool may become charred by heat from the pipe and finally ignited, though this can hardly happen, even on high-pressure pipes, when the thickness of fire-proof material (asbestos, magnesia, or whatever it may be) is as great as one inch. The second and most serious risk is from the presence in shops or mills of the long tubes of wool, dry as tinder, often connecting one room with another, and ready to flash at the slightest rise in the already too great temperature. Canvas jackets on the covers should be fire-proof. The efficiency of wools is high as non-conductors, but not higher than any other perfectly safe covers. If the wool is separated by about one inch of fire-proof material from the pipe, it is not kept so hot and dry, and the risks from outside ignition is less; but the practice of many engineers of wrapping hair-felt outside of a sectional cover is not advised. The saving due to this practice is indicated in Table E.

The following assumptions have been made in computing the Tables D, E and F. First, that all the covers cost £5 per one hundred square feet, applied. This is a high figure, perhaps too high, yet it is not far from the list price of several makers, and any attempt to get a definite price from them—revealed a maze of discounts and double discounts and flexible price-lists too intricate for an uninitiated mind to travel. In case the saving due to a cover, which costs £4 instead of £5, is desired, the simple

addition to the final saving of the £1 difference makes the necessary correction.

Secondly, by the advice of the makers, the assumption is made that the cost is not nearly proportional to the thickness. As the thicker coverings are not now made in great quantities, the actual cost of their manufacture is uncertain.

Inspection of Table E shows the saving due to the use of hair-felt outside a standard magnesia cover.

In five years one hundred square feet of hair-felt saves £1·40 more than its cost, and in ten years it saves £4 above its cost.

The further saving due to a second inch outside the first is £1·60 in ten years. Of course the well-known tendency of hair-felt to deteriorate should be considered.

In the case of Nonpareil cork, increasing the thickness from one to two inches raises the cost from about £5 to £7 per one hundred square feet, and increases the net saving in five years by £2 and by £6 in ten years. In other words, the second inch of material in use about pays for itself in two years, while the first pays for itself in about one year. The third inch does not increase the saving even in ten years. The second inch, therefore, more than pays for interest and depreciation, while the third fails to do this.

In the case of the asbestos fire board, a second inch in thickness causes a saving of £4 in ten years, the third and fourth inches showing a loss.

TABLE E.

VARIATIONS IN THICKNESS, ETC.

Specimen.	Saving in B.Th.U. per sq. ft. per minute.	Saving in £ per 100 sq. ft. per year.	Net Saving.				Approximate Cost.
			1 Year.	2 Years.	5 Years.	10 Years.	
Magnesia $1\frac{3}{8}$ in. thick	11·62	£7·55	£1·55	£9·10	£31·8	£69·4	£6·00
Magnesia $1\frac{3}{8}$ in. thick and 1 in. of Hair-Felt	12·38	8·04	1·04	9·09	33·2	73·4	7·00
Magnesia $1\frac{3}{8}$ in. thick and 2 in. of Hair-Felt	12·77	8·30	0·30	8·60	33·4	75·0	8·00
Nonpareil Cork :							
1 inch	11·64	7·56	2·56	10·00	32·8	70·6	5·00
2 inch	12·84	8·35	—	9·70	34·8	76·6	7·00
3 inch	12·94	8·41	—	6·82	32·0	72·0	10·00
Fire Board :							
1 inch	10·54	6·84	1·84	8·68	29·2	63·4	5·00
2 inch	11·48	7·45	0·45	7·90	30·2	67·2	7·00
3 inch	11·70	7·60	2·40	5·20	28·0	66·0	10·00
4 inch	11·83	7·68	5·32	2·36	25·4	63·8	13·00

In general it may be said, therefore, that if five years is the length of life of a cover, one inch is the most economical thickness, while a cover which has a life of ten years may to advantage be made two inches thick.

In view of the custom which prevails to some extent of wrapping asbestos paper round a pipe and surrounding the whole with hair-felt, tests were made as to the temperature of the bounding line of the asbestos paper and hair-felt, using a Le Chatelier thermo-electric pyrometer for this purpose. The different samples of asbestos paper give widely varying results, but a general idea of the protection afforded by the paper may be had from Table F.

TABLE F.

PROTECTION AFFORDED BY ASBESTOS PAPER. PIPE AT 200 POUNDS PRESSURE.

Thickness of Asbestos Paper.	Temperature of Pipe.	Temperature of Inside of Hair-Felt.	Pressure Corresponding to the Temperature of the Inside of the Hair-Felt.
$\frac{1}{64}$ inch	384·7° Fahr.	356° Fahr.	146 pounds
$\frac{1}{32}$,,	385·0° ,,	329° ,,	102 ,,
$\frac{1}{16}$,,	384·6° ,,	302° ,,	70 ,,
$\frac{1}{8}$,,	384·7° ,,	266° ,,	39 ,,

Attention being called to the varying loss from bare pipes when their surfaces were in varying conditions as regard rust, dirt, paint, etc., a few brief tests to show any large variation which might occur

from the loss from bare pipe, viz. 13·84 B.Th.U. per square feet per minute, are shown in Table G.

TABLE G.

LOSS OF HEAT AT 200 LB. FROM BARE PIPE.

Condition of Specimen.	B.Th.U. loss per sq. ft. per minute.
New pipe	11·96
Fair condition	13·84
Rusty and black	14·20
Cleaned with caustic potash inside and out .	13·85
Painted dull white	14·30
Painted glossy white	12·02
Cleaned with potash again . . .	13·84
Coated with cylinder oil	13·90
Painted dull black	14·40
Painted glossy black	12·10

The rate of heat loss from a bare pipe is also affected by the air circulation and the temperature of the surrounding bodies. A few tests were made to indicate the magnitude of the errors likely to be caused by variation in these conditions, and a brief examination of some of the results may be interesting. They are given in Table H.

Table I shows the varying loss from a bare pipe with the change in pressure.

A very thorough test was made of the common method of judging a pipe cover by the sensation of warmth given the hand on touching it, and nothing too harsh can be said of this practice. The sensation is dependent to such an extent upon the

nature of the surface that it fails utterly to give any idea of the actual temperature.

TABLE H.

EFFECT OF SURROUNDINGS.

Condition and Position of Pipe.	B.Th.U. lost per sq. ft. per minute at 200 pounds.
1. Standard condition; hung in centre of room	13·84
2. Near brick wall, between windows . .	14·26
3. Hung horizontally in centre of room .	12·06
4. Vertical 10-inch pipe { 36 inches long .	13·48
4. Vertical 10-inch pipe { 18 ,, ,, .	14·42
5. Vertical 18-inch. long { 10-inch diameter .	14·42
5. Vertical 18-inch. long { 4-inch ,, .	15·20
6. 4-inch diameter in draft from electric fan .	20·10

TABLE I.

VARIATION OF HEAT LOSS WITH PRESSURE.

Pressure. Lb.	Bare Pipe Loss B.Th.U. per sq. ft. per Minute.
340	15·97
200	13·84
100	8·92
80	8·04
60	7·00
40	5·74

The ease of removal for repairs or alterations makes the sectional cover better than plastic for some work, but there is much pipe surface which might be covered securely with plastic, where a sectional cover is soon ruined by vibration. Of course, the plastic covers offer no possibility of

leaky joints and long cracks. It should be borne in mind that in most cases about 20 per cent. of the entire surface to be covered is irregular, and must be covered by plastic or fittings. It will be well for prospective purchasers of pipe cover to see to it that their contracts call for fittings and plastic of as high an efficiency as the sectional cover shows.

TABLE K.

MISCELLANEOUS SUBSTANCES.

Specimen.	B.Th.U. per sq. ft. per minute at 200 pounds.	Saving in one year per 100 sq. ft. pipe.
Box A:		
1. With sand	3·18	£6·92
2. With cork, powdered. .	1·75	7·88
3. With cork and infusorial earth	1·90	7·78
4. With sawdust . . .	2·15	7·58
5. With charcoal . . .	2·00	7·70
6. With ashes . . .	2·46	7·38
Brick wall 4 inches thick . .	5·17	5·76
Pine wood 1 inch thick . . .	3·56	6·76
Hair-Felt 1 inch thick . . .	2·51	7·36
Cabot's seaweed quilt 1 inch thick.	2·78	7·18
Spruce 1 inch thick . . .	3·40	6·76
,, 2 inches thick . . .	2·31	7·30
,, 3 inches thick . . .	2·02	7·70
Oak 1 inch thick	3·65	6·62
Hard pine 1 inch thick . . .	3·72	6·58
Eider-down 1 inch thick loose .	*1·90 to 2·70	—
,, 1 inch thick tightly packed	*1·70 to 1·80	—

* Variable.

Table K gives some figures concerning a considerable number of samples of non-conducting material,

not, perhaps, classed as pipe covers, but used for heat insulation, which may be of interest.

The box A, referred to in the table, is a $\frac{7}{8}$-inch pine box, large enough to surround the pipe, leaving a one-inch minimum space at its four sides. In it were tested several materials, which are used in this way for steam and cold storage insulation.

CHAPTER XVI

Weights of Pipe

THE weight of a pipe is usually found by multiplying the length in feet by the weight of a foot length of pipe, and adding two flanges. Excellent tables of weights of pipes and various junction pieces are published by several firms who supply pipes. For work not using the pipes of any special maker the weight of iron may be taken as 20 pounds per square foot for material $\frac{1}{2}$-inch thick, and *pro rata*.

For steel pipes from $\frac{1}{8}$ to 1 inch thick, and from 10 I.W.G. to 1 I.W.G. the tables of the Mannesmann Company are very full.

To calculate the weight of a tube multiply together its length in feet and its mean diameter in inches, and the number $10{\cdot}6 \times$ thickness. Thus a pipe $6\frac{1}{4}$ *external* diameter and $\frac{1}{4}$ thick will weigh per 1 foot long $6 \times 10{\cdot}6 \times 0{\cdot}25$. Approximately, for ordinary steam pipes, 10 times the *external* diameter $\times$ thickness $=$ weight per foot. Thus for a $6\frac{1}{4}$ external diameter pipe $\frac{1}{4}$ thick we have $6{\cdot}25 \times 10 \times 0{\cdot}25 = 15{\cdot}6$. The figure in the Mannesmann tables is $15{\cdot}9$. The rule gives results about 2 per cent. light for small thin pipes up to 3 per cent. heavy for larger heavy pipes, but it is very close to truth for pipes of

ordinary use, and crosses the line of plus and minus error at 8 inches external diameter. Cast iron weighs about 6 per cent. less than steel, or 9·375 pounds per square foot $\frac{1}{4}$ inch thick. The following weights per superficial foot 1 inch thick will be useful—

	lb.		lb.
Cast Iron . . .	37·50	Copper . . .	46·2
Wrought Iron . .	40·42	Brass . . .	43·3
Steel . . .	40·82	Lead . . .	59·5

Thus wrought iron is 1 per cent. heavier than the rule above allows for, and steel is another 1 per cent. heavier.

Brazed copper tubes weigh somewhat more than solid drawn. The specific gravity of tough copper at 60°F. is 8·8917, or 0·3229 lb. per cubic inch, or practically 3 cubic inches to the pound.

	Specific Gravity.	Weight per cubic inch.
Aluminium	2·56	·0926
Brass, from	8·82	·3194
,, to	7·82	·2828
Gunmetal	8·70	·3147
Copper	8·69	·3146
,, drawn . . .	8·88	·3212
,, pipe	8·89	·3229
Iron, Cast	7·21	·2607
,, wrought bar . . .	7·79	·2817
,, rolled plate . . .	7·70	·2787
Lead, Cast	11·35	·4106
,, rolled	11·39	·4119
Nickel	8·80	·3183
Steel plate	7·80	·2823
Zinc, rolled	7·19	·2600
Tin	7·29	·2637

Useful tables for the weight of copper and iron, steel and cast-iron pipes will be found in Kempe's *Year-Book*, or other pocket-books and manufacturers' catalogues, and are therefore not given here.

JNO. SPENCER'S, LTD., STANDARD DIMENSIONS OF TUBULAR IRON & STEEL FLANGED BENDS (FIG. 60).

D		A	B	C
Bore of Pipe. Inches.		Radius at Centre. Inches.	Length Straight. Inches.	Centre to Flange Face. Inches.
$\frac{1}{2}$		2	$2\frac{1}{2}$	$4\frac{1}{2}$
$\frac{3}{4}$		$2\frac{1}{2}$	$2\frac{1}{2}$	5
1		3	3	6
$1\frac{1}{4}$		$3\frac{3}{4}$	3	$6\frac{3}{4}$
$1\frac{1}{2}$		$4\frac{1}{2}$	3	$7\frac{1}{2}$
$1\frac{3}{4}$		$5\frac{1}{4}$	$3\frac{1}{2}$	$8\frac{3}{4}$
2		6	$3\frac{1}{2}$	$9\frac{1}{2}$
$2\frac{1}{2}$	A=3 D	$7\frac{1}{2}$	4	$11\frac{1}{2}$
3		9	4	13
$3\frac{1}{2}$		$10\frac{1}{2}$	5	$15\frac{1}{2}$
4		12	5	17
$4\frac{1}{2}$		$13\frac{1}{2}$	6	$19\frac{1}{2}$
5		15	6	21
6		18	7	25
7		$24\frac{1}{2}$	7	$31\frac{1}{2}$
8	A=$3\frac{1}{2}$ D	28	8	36
9		$31\frac{1}{2}$	8	$39\frac{1}{2}$
10		40	9	49
11	A=4 D	44	9	53
12		48	10	58
13		$58\frac{1}{2}$	11	$69\frac{1}{2}$
14	A=$4\frac{1}{2}$ D	63	11	74
15		$67\frac{1}{2}$	12	$79\frac{1}{2}$
16		80	13	93
17	A=5 D	85	14	99
18		90	14	104
19	A=$5\frac{1}{2}$ D	$104\frac{1}{2}$	15	$119\frac{1}{2}$
20		110	16	126

Bolt weights may be found in any pocket book; sufficient to remember that a yard of iron or steel rod 1 inch square weighs 10 pounds per yard, and 7·85 pounds of 1 inch diameter, or for any size, its weight per yard is $D^2 \times 10$ pounds if square, and $D^2 \times 7{\cdot}85$ if round.

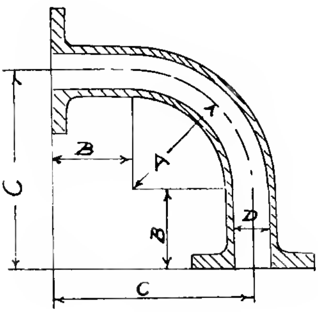


FIG. 60.

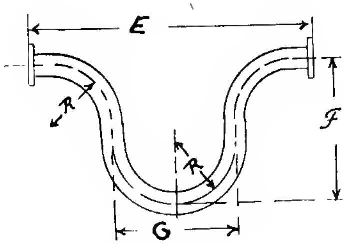


FIG. 61.

DIMENSIONS OF BENDS (JNO. SPENCER, LTD.)

(FIG. 61.)

Bore.	E		F		G		R		
in.	ft.	in.	ft.	in.	ft.	in.	ft.	in.	
1	2	0	1	0	0	6	0	3	
2	3	0	1	6	1	0	0	6	
3	4	0	2	0	1	6	0	9	
4	5	0	2	6	2	0	1	0	
5	6	0	3	0	2	6	1	3	
6	7	0	3	6	3	0	1	6	
7	8	0	4	0	3	6	1	9	
8	9	0	4	6	4	0	2	0	
9	10	0	5	0	4	6	2	3	
10	11	0	5	6	5	0	2	6	
11	12	0	6	0	5	6	2	9	To be in 3 pieces.
12	13	0	6	6	6	0	3	0	

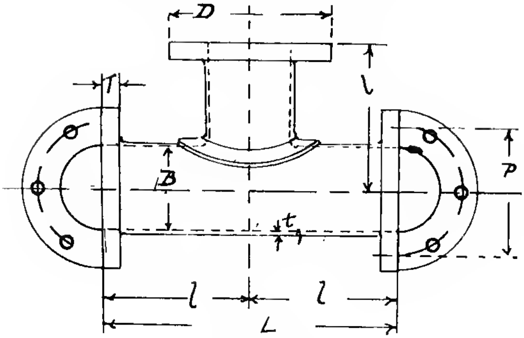


FIG. 62.

FOR SOLID WELDED & RIVETED FLANGES (FIG. 62)

B	D	N	d_1	P	d	T	t	l	L
in.	in.		in.	in.	in.	in.	in.	in.	in.
4	9	6	3/4	7 1/4	7/8	3/4	1/4	8	16
5	10 1/2	8	3/4	8 3/4	7/8	7/8	1/4	10	20
6	12	8	3/4	10	7/8	1	1/4	11	22
7	13 1/2	8	3/4	11 1/2	7/8	1	1/4	12	24
8	15	8	7/8	12 3/4	1	1 1/8	5/16	13	26
9	16	10	7/8	13 3/4	1	1 1/8	5/16	14	28
10	17	10	7/8	14 3/4	1	1 1/8	5/16	15	30
11	18	12	7/8	15 3/4	1	1 1/4	5/16	16	32
12	19	12	7/8	16 3/4	1	1 1/4	3/8	17	34
13	21	14	1	18	1 1/8	1 1/4	3/8	18	36
14	22	14	1	19	1 1/8	1 3/8	3/8	19	38
15	23	16	1	20	1 1/8	1 3/8	3/8	20	40
16	24	16	1	21	1 1/8	1 3/8	7/16	21	42
17	25	18	1	22	1 1/8	1 3/8	7/16	22	44
18	26	18	1	23 3/4	1 1/8	1 1/2	7/16	23	46
19	27	20	1	24 3/4	1 1/8	1 1/2	7/16	24	48
20	28 1/2	20	1 1/8	26	1 1/4	1 3/4	7/16	25	50
21	30	20	1 1/4	27 1/4	1 3/8	1 3/4	7/16	26	52
22	31 1/4	20	1 1/4	28 1/2	1 3/8	2	7/16	27	54
23	32 1/2	20	1 1/4	29 1/2	1 3/8	2	7/16	28	56
24	33 1/2	20	1 1/4	30 1/2	1 3/8	2	7/16	29	58

d_1 Dia. of Bolt. d Dia. of Holes. N No. of Holes.

CHAPTER XVII

The Kinetic Theory of Gases in its Relation to the Flow of Steam

THE kinetic theory of gases is based on the assumption that the molecules composing a gas are small bodies possessed of motion by virtue of heat. All the properties of gases are explicable by this theory. Pressure, for example, consists of the impact of the molecules on the containing boundaries of the vessel. Density variation is brought about by a variation in the number of molecules in a given space. Pressure is varied by adding or subtracting heat, because, by so doing, the impact velocity of the molecules is made greater or less. The molecules are assumed almost perfectly elastic, so that they rebound from a surface at nearly the velocity with which they strike it. The molecules obey the first law of motion, for, being in motion, they continue to move at the same velocity in a straight line unless acted upon by some force external to themselves. In any body of gas, therefore, the countless molecules are moving in ceaseless collision with each other and the containing boundaries, and the mean result is a steady pressure and temperature of the mass.

If an opening be made in the restraining boundary, the truth of the hypothesis is made evident, for the molecules, which at the time are approaching and are near to the opening, rush out at once and overcome the opposing molecules of the air outside, or *vice versa*. The outrush or inrush continues until a balance is effected. Until the flowing stream falls to the pressure of the surrounding medium the molecules will not even tend to move in parallel straight lines. There will be transverse motion and the stream will widen and thicken, or expand, until the strength of bombardment of the surrounding medium is equalized.

When a gas flows into a vacuum there are no opposing molecules in front of it to drive back, and the velocity of flow is obviously the inherent mean velocity of the molecules themselves at the then temperature.

Clausius calculated this velocity at 0°C. =32°F. as follows :—

	Density $= d$	Metres per second.	Feet per second.
Air	14.5	485	1591
Oxygen	16	461	1513
Hydrogen	1	1844	6050
Nitrogen	14	492	1618
Steam	9	615	2017
Carbon Monoxide	14	493	1618
Carbon Dioxide	22	392	1286

The velocity in metres per second is $1844 \times \sqrt{d}$

where d is the density relative to hydrogen, for which $d = 1$.

The velocity varies inversely as the square root of the density, and it varies proportionately with the square root of the absolute temperature, as naturally follows from Boyle's law.

At 0° Centigrade, therefore, the density of steam being 9, its velocity will be 615 metres per second = 2,017 feet per second.

This being the velocity of the steam-molecule, represents also the velocity of flow which it tends to achieve into a perfect vacuum. Excepting so far as pressure in the case of saturated steam has its own particular temperature, the pressure of a gas is no measure of its molecular velocity. We can therefore calculate the velocity of flow of steam into a vacuum if we know its temperature, and the following table shows the results calculated for a few cases from the datum 615 metres per second at 0°C.

Experiment has shown that steam flowing from one pressure to another not greater than 58 per cent. of the initial pressure, attains a maximum of flow whence is deduced a rule that $W = \frac{6\,A.P.}{7}$, where

W = weight of flow per minute in pounds.
P = absolute pressure in pounds.
A = area of orifice in square inches.

Calculated out for 100 lb. absolute pressure, the outflow per second through an area of one square

Pressure. Absolute. lb.	Temperature.		Absolute Temperature.		Velocity.	
	C°	F°	C°	F°	Metres per sec.	Feet per sec.
0·207	12·34	54·21	285·34	513·21	629	2064
0·453	24·89	76·80	297·89	535·80	642	2106
0·698	32·35	90·24	305·35	549·24	649	2129
0·944	37·78	100·05	310·78	559·05	656	2152
1·189	42·13	107·84	315·13	566·84	660	2165
1·435	45·74	114·34	318·74	573·34	664	2179
1·680	48·85	119·94	321·85	578·94	667	2188
1·926	51·59	124·89	324·59	583·89	670	2198
2·172	54·06	129·31	327·06	588·31	673	2208
2·427	56·28	133·32	329·28	592·32	675	2214
4·873	71·80	161·25	344·80	620·25	691	2267
5·856	76·14	169·07	349·14	628·07	695	2280
6·838	79·92	175·87	352·92	634·87	699	2293
14·697	100·0	212·00	373·00	671·00	718	2356
50·000	138·3	280·90	411·30	739·90	755	2477
100·00	164·2	327·63	437·20	786·63	778	2553
150·00	181·2	358·22	454·20	817·22	793	2602
200·00	194·20	381·64	467·20	840·64	800	2625
250·00	203·00	401·10	476·00	860·10	812	2664
300·00	214·14	417·50	487·14	876·50	821	2694
—	260·0	500·00	533·00	959·00	859	2818
—	315·5	600·00	588·55	1059·00	903	2961

foot would be 205·7 pounds, whereas the tabular number 2,553 cubic feet reduced to pounds gives 586·4 as the weight that should apparently pass by the kinetic theory. The discrepancy probably arises because molecules of an enclosed gas are moving in every direction relative to an orifice in the wall of the vessel, and those molecules travelling straight for the outlet are hindered by those moving along the other two space dimensions. To pass a full quantity through an opening, the approach to that opening should be of the correct tapering form and so should also be the outlet end. The effect is, in

fact, simply the commonly recognized *vena contracta* effect. But as much steam will flow into a pressure of one-half the initial pressure as into a vacuum.

If it were possible to persuade all the molecules of a flowing jet of steam to move together in parallel lines then by directing the stream upon the suitably shaped vanes of a turbine moving at a velocity one-half that of the steam, the steam molecules would drop from the turbine deprived of all energy, or movement, or heat, and the efficiency of the turbine would be 100 per cent. We know that a heat engine cannot have an efficiency of more than $E = \frac{T_1 - T_2}{T_1}$ where T_1 and T_2 are the upper and lower absolute temperatures, and that the assumed molecular movement is not possible, but this idea is at the bottom of the steam turbine, in which the action of the steam is obtained by allowing its molecular kinetic energy to manifest itself as mechanical kinetic energy—a distinction without a difference, for all steam energy is really kinetic.

Though the mean molecular velocity of steam may fall short of 3,000 ft. per second in all practical cases, certain writers refer to a possible outflow velocity as high as 5,000 ft. per second. Apparently this velocity could only occur where the flowing molecules were assisted by the energy of those behind, which would be much cooled by the operation, losing as much velocity as the outgoing molecules had gained over and above the mean value.

This hardly seems possible, and velocities above those in the table seem hardly probable.

Those who are interested in the subject cannot do better than study Meyer's work on the *Kinetic Theory of Gases*.[1]

The theory is certainly an aid in the comprehension of the behaviour of flowing steam, and if clearly grasped, it will help to explain why the efficiency of a heat engine is so small.

[1] *The Kinetic Theory of Gases*, by O. E. Meyer. Translated by Robert Baynes, M.A. Longmans and Co.

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